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26,853 — 26,930

THE BRITISH SHIP RESEARCH ASSOCIATION

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THE RESEARCH COUNCIL

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TABLE OF CONTENTS

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ABSTRACTS *from Current Technical Literature*

The following Abstracts purport to be fair summaries of the articles, but the Association does not accept responsibility for statements made in the originals, nor does it necessarily agree with their contents.

The standard form of reference to the source of each Abstract is: Title of Periodical or Publication (abbreviated according to the list on pp. 3–19 of B.S.R.A. Journal for January 1968), volume number (in heavy type), year, and page number, followed by the date of issue where appropriate. The length of the article and other bibliographical details are also indicated.

SHIP RESISTANCE AND FLUID MOTION

26,853 **An Analysis of Ship Model Resistance into Viscous and Wave Components. Parts I and II.** HUGHES, G. *Trans. R.I.N.A.*, **108** (1966), p. 289 (July) [9 pp., 7 ref., 5 graphs, 3 diag.; and Discussion, 6 pp., 2 ref., 1 tab., 1 graph]

This paper was read in February 1966.

Part I of the paper is a general discussion of the scale-effect problem, leading to the reasons for making the comprehensive analysis of N.P.L. resistance data which is currently in progress, and for adopting the chosen procedure. The method of analysis into viscous and wave components is discussed and a brief outline of the manner in which the results will be recorded and co-ordinated is given. The aim of the investigation, which will include all types of hull form, is to discover a formula giving the resistance of any reasonable form when some of its parameters are known, and to show how variations in form can affect viscous and wave-making resistance. It is assumed that the wave resistance varies as the fourth power of the speed, and that there is no change caused by viscosity in the wave resistance.

Part II gives preliminary results for the viscous and wave resistance coefficients based on the analysis of about 150 model C curves. This analysis is based on only two main parameters: the block coefficient and the (Displacement)$^{1/3}$/Length ratio. It is shown that the total-resistance coefficients for all classes of vessel within normal operational speed ranges may be expressed with fair accuracy by a general formula involving Reynolds number and the fourth power of the speed–displacement constant K.

See also following Abstract.

26,854 **An Analysis of Ship Model Resistance into Viscous and Wave Components. Part III—The Viscous Resistance Coefficient.** HUGHES, G. *R.I.N.A., Paper No. W8* (1967), *issued for written discussion* [7 pp., 1 ref., 7 graphs]

For Parts I and II of this work, see preceding Abstract.

In Part III the values of the viscous resistance coefficient obtained from the analysis of nearly 400 model total-resistance results are presented. The dependence of this coefficient on block coefficient and on wetted surface area coefficient is shown for a wide range of ship designs. The results also cover wide ranges of breadth/draught ratio, midship-section

coefficient, and longitudinal position of the centre of buoyancy, and it is seen that variation of these factors has relatively little effect on the values of the viscous resistance coefficient. This work enables values of this coefficient to be estimated for a large variety of hull forms with sufficient accuracy for scaling within the model range and for extrapolation to full scale.

26,855 **A New Turbulent Friction Formulation Based on a Re-appraisal of Hughes' Results.** GADD, G. E. *R.I.N.A., Paper No. W7* (1967), *issued for written discussion* [13 pp., 12 ref., 2 tab., 16 graphs, 5 diag.]

Hughes' experimental results (see Abstract No. 8874, May 1954) for the turbulent skin-friction coefficient on narrow rectangular flat surfaces are re-examined in the light of published work on the " edge effect ", and of N.P.L. experiments (fully described in the present paper) in which the friction distributions across glass plates were measured by arrays of Preston tubes (plate width and immersion being varied). A new basic friction line for surfaces of infinite width (aspect ratio) is proposed. This line is higher and flatter at high Reynolds numbers than the one proposed by Hughes; it is expressed by the equation $C_F = 0{\cdot}0113/(\log_{10}R_L - 3{\cdot}7)^{1{\cdot}15}$, R_L being the length-based Reynolds number. It is thought to be approximately valid over the R_L range 10^6 to 10^8.

26,856 **Some Experiments on the Incremental Resistance due to Rolling with and without Bilge Keels and Vanes.** GADD, G. E. *Trans. R.I.N.A.*, **109** (1967), p. 327 (July) [5 pp., 3 ref., 4 tab., 6 graphs, 5 diag., 2 phot.; and Discussion: 4 pp., 3 ref., 3 graphs]

The incremental resistance due to rolling was measured for a 10-ft model of a trawler form, run at about the ship self-propulsion condition and fitted with forced-rolling equipment. When bilge keels were fitted it was found to be less than about 20% of the calm-water resistance for roll angles up to 10° each side of the vertical. Triangular bilge vanes (several arrays of which were tested) produced about the same incremental resistance as keels. Their roll-damping characteristics agreed fairly well with theoretical predictions, i.e. vanes (preferably disposed in two rows on each bilge) seem better than keels for ships of relatively slow rolling period and relatively high speed.

See also Abstract No. 22,940 (May 1965).

26,857 **Some Research on the Hydrodynamics of Catamarans and Multi-Hulled Vessels in Calm Water.** EVEREST, J. T. *Trans. N.E.C. Instn E. Shipb.*, **84** (1968), p. 129 (May) [20 pp., 10 ref., 2 tab., 27 graphs, 6 phot.]

The Author presents a summary of the work done during the initial stages of an investigation, at the N.P.L., into the calm-water performance of multi-hull vessels. See also Abstract No. 26,039 (Jan. 1968).

A simple method is outlined for estimating inter-hull wavemaking interference from experimental knowledge of the wave pattern of a single hull. It is an extension of Eggers' procedures—see Abstracts No. 18,433 (June 1962), and 25,195 (Apr. 1967). The limitations of representing the basic hull completely theoretically are avoided by this approach, and a range of different component-hull designs can be compared analytically; a computer program has been written which simplifies the analysis.

Interaction between the hulls is shown to have an important effect on

viscous resistance. In general, measured wave-pattern drag agrees well with predictions, but the prediction of total drag at small hull-separations is less exact because of lack of data about viscous interaction. The distribution of wave energy between transverse waves and divergent waves is shown to be very important. It seems that, for catamarans, the most beneficial interference effects occur when the component hulls have strong divergent wave systems; strong transverse waves result in adverse interference. The optimum speed range for catamarans is, roughly, $F_n = 0{\cdot}3$ to $0{\cdot}4$; reductions in wave resistance of over 50% (compared with twice the single-hull value) can be achieved with careful design. For trimarans (three identical hulls), a 60% reduction in wave resistance can be achieved over a wide speed range, provided the centre hull has forward stagger.

The detailed information is given under the headings:—

1. Prediction of Catamaran Wavemaking Resistance.
   (i) Method of Estimation.
   (ii) Further Comments on the Proposed Method.
   (iii) Results of the Prediction Method (mathematical hull form applied in catamaran configuration and in trimaran configuration; conventional hull form applied in catamaran configuration).
   (iv) Some Thoughts on the Wave Cancellation.
2. Comparison of Wavemaking Predictions with Experimental Results (mathematical hull form applied in catamaran configuration; conventional hull form applied in catamaran configuration).
3. Viscous Interference between Hulls.
4. Comparison with the Equivalent Single-Hulled Vessel.
5. Conclusions.

Appendix 1—Outline of the Theory used to Determine Wave-Pattern Data.

Appendix 2—Further Comments on the Hydrodynamics of Multi-Hulled Ships in Comparison with the Equivalent Single-Hulled Vessel (catamaran ships; trimaran ships).

26,858 **Ship Resistance and Propulsion—Model/Ship Correlation Methods** (in German). CHIRILA, J. V. *Schiff u. Hafen*, **19** (1967), p. 881 (Dec.) [10 pp., 29 ref., 4 tab., 7 graphs, 2 diag., 1 phot.]

This paper was read at the Nov. 1967 Meeting of the Schiffbautechnische Gesellschaft, and is accompanied by a brief account of the subsequent discussion. The paper is based on a five-volume unpublished work by the Author on resistance and propulsion testing.

The Author first discusses the importance of shipboard thrust measurement, explains its principles and practice, and describes the Simplex hydraulic thrust-measurement appliance, which can measure mean thrust and thrust fluctuations. The construction and operation of this device are explained with the aid of drawings; it can incorporate recording and remote-reading equipment.

The rest of the article is concerned with model/ship correlation, under

the headings: (i) Methods of evaluating sea-trials results. (ii) Calculation of full-scale propulsion characteristics from thrust measurements. (iii) Economic ship-speed. (iv) Resistance correlation-factor. (v) Establishing a new correlation line for viscous resistance. (vi) Viscous resistance coefficient. (vii) Determination of the correlation gradient for the specific total-resistance coefficient. (viii) Model/ship correlation for the r.p.m./wake relationship. (ix) Correlation for thrust-deduction fraction and thrust/torque relationship. (x) Relative rotative efficiency.

A comparison is made between correlation methods employing the frictional resistance coefficients of Froude, Schoenherr, ITTC 57, and Schultz-Grunow. These methods do not give satisfactory results, and the Author therefore develops an improved correlation expression for viscous resistance. This expression takes into account the effect of hull form.

The Author also develops a new expression for wake-fraction correlation. This is dependent on hull form and on the viscous-resistance coefficient; some examples are given in which shipboard thrust measurements are used to verify the expression. Basic data are derived, from shipboard thrust measurements, for the correlation of the thrust-deduction fraction.

26,859 **An Assessment of Some Experimental Methods for Determining the Wavemaking Characteristics of a Ship Form.** EGGERS, K. W. H., SHARMA, S. D., and WARD, L. W. *Soc. N.A.M.E., paper presented at Annual Meeting*, 15-18 Nov. 1967 [33 pp., 66 ref., 3 tab., 11 graphs, 7 diag., 1 phot.]

The experimental determination of wave spectra and the corresponding wave resistance from measurements of the wave pattern behind a model while being towed in a conventional model tank has received considerable attention in recent years, and a number of possible methods have been proposed. Successful development of such techniques promises major advances in the improvement of wave-resistance theory and in increased effectiveness of the model tank as a tool for the investigation and improvement of hull forms. During the academic year 1965 66 the Authors carried out tests and analytical studies, including computer evaluations, in Germany using several of the methods applied to the same model, in order to obtain an assessment of their relative merits as well as to test the consistency of the basic approach. The paper presents the hydrodynamic theory and the mathematical model on which all the methods are based. Results of the experimental and computational investigations are given and compared. Conclusions are drawn as to the validity, ease of performance, and adaptability to existing tanks of each of the methods. The results are generally encouraging and indicate that a consistent quantity is being measured. An extensive bibliography is included.

26,860 **Tests with a Deeply-Submerged Double-Model of a Cargo Ship in the Rotating-Arm Basin** (in German) SEILER, K.-D. *Schiffbauforschung*, 7 (1968), p. 37 (Special Issue) [15 pp., 4 ref., 2 tab., 17 graphs, 3 diag., 4 phot.]

This paper was presented at the Schiffbautechnische Herbsttagung held in Rostock in Oct. 1967.

When the steering characteristics of a ship are being predicted in the

design stage, one source of inaccuracy in the calculations is the omission to take into consideration the effects of the free water surface on the forces and moments imposed on the hull. These effects can be investigated by tests on an immersed double-model and, *inter alia*, comparing the results with those from tests on a normal surface-model. The Author gives an account of such tests in which a 1 : 71 double-model of the Warnowwerft Type IV 10,000-ton twin-screw cargo-ship was used. The tests were run in the rotating-arm basin at Rostock University, with the model submerged to 1·5 m in a channel depth of 3 m (9·8 ft). The account includes a detailed description of the experimental equipment and techniques.

The results are presented in curves and are discussed. It was found that, at low Froude numbers, the presence of the water surface diminished the transverse force by about 10 to 20%, and that its effect on the moment coefficient was very small. The influence of wave formation on the forces and yawing moment, at small angles of yaw, does not begin until a Froude number of about 0·23 is reached. In the speed-range investigated, which is applicable to normal cargo-ships and passenger-ships, the transverse-force and moment coefficients are independent of Reynolds number. Fitted polynomials can be used to approximate the experimental results for the calculation of steered ship motion. For small angles of yaw, the calculated transverse-force and moment coefficients agree well with the experimental results.

PROPELLERS AND PROPULSION

(*See also Abstracts No.* 26,858 *and* 26,903)

26,861 **Propeller Design and Analysis by Lifting Surface Theory.** MURRAY, M. T. *Int. Shipbuild. Progress*, **14** (1967), p. 433 (Dec.) [18 pp., 7 ref., 4 tab., 8 graphs, 4 diag.]

The Author derives and uses the basic equations of lifting-surface theory to provide methods for the design and evaluation of propellers. For the design calculation, the method requires as data the number of blades and the blade outline and thickness distribution, the axisymmetric wake inflow pattern and advance ratio, and the required thrust together with the shape of the circulation distribution over the blade surface; the method will then yield a value of the torque required, and also the blade pitch-angle distribution and the distribution of camber, cavitation number, and actual circulation values over the blade surface. For the evaluation calculation, the method requires full details of the propeller and operating conditions, and will yield thrust and torque values and also surface distributions of circulation value and cavitation number.

The treatment includes the case of a pair of contra-rotating propellers and the similar case of a rotor stator combination. The method will treat straight or skewed propellers.

The numerical techniques used in the calculations are described. Some information is also given about the English Electric KD F9 computer program which was written to carry out the computation. Numerical examples are presented.

In his conclusions, the Author states that the work described has been checked, where possible, by computing examples for which theoretical results obtained by other workers are already available. Some propellers

have already been designed using the computer program described, and their experimental evaluation is awaited. Other propellers are at present being designed for manufacture and testing, and it is intended to analyse a wide variety which have already been tested. The theory, and the associated computer program, should be developed to take into account change in pitch of the streamlines downstream, contraction of the wake, and also hub size and configuration. An extension of the calculations to include ducted propellers is also suggested.

26,862 **Open-Water Propeller Tests at Different Immersions** (in German). GUTSCHE, F. *Schiffbauforschung*, **7** (1968), p. 32 (Special Issue) [4 pp., 1 ref., 25 graphs]

This paper, presented at the Schiffbautechnische Herbsttagung held in Rostock in Oct. 1967, is complementary to Publication No. 45 of the Schiffbau-Versuchsanstalt, Berlin (Potsdam), which was summarised in Abstract No. 26,382 (May 1968).

In the work described in that publication, the intention was to establish influence factors which would enable the results of the model tests (on the effect of depth of immersion on the thrust and efficiency of various propellers) to be used in predictions for any propeller. That intention was not fully realised; this is attributed, in the present paper, to the fact that air drawing to the suction side of the blades depends not only on Froude number but also, to a significant extent, on blade section and on Reynolds number. With these considerations in mind, the Author makes some additional plots of results given in Publication No. 45 of the SVA, and devises a suitable interpolation procedure.

26,863 **Some Effects of Variation in Blade Area, Blade Outline, and Boss Diameter on Model Screw Performance.** O'BRIEN, T. P. *Trans. N.E.C. Instn E. Shipb.*, **84** (1968), p. 149 (May) [12 pp., 8 ref., 18 tab., 10 graphs, 11 diag.]

When making ship-propeller performance predictions based on the results of propulsion tests on model standard-series propellers, correction factors are used to take into account departures from the standard. Some results of N.P.L. investigations on the effects of variation in blade-section shape, blade thickness, and number of blades have previously been published (see, for example, Abstract No. 23,299, Aug. 1965). The present paper gives an account of a continuation of that work. The results, which are discussed, include comparisons of performance under non-cavitating and cavitating conditions, based on open-water and water-tunnel experiments. Correction factors are derived and design data are given which enable propellers of different blade area, blade outline, and boss diameter to be designed, and comparative estimates of their performance to be made, using data for a standard propeller as a basis.

26,864 **An Investigation of the Scale Effect on Self-Propulsion Factors.** BENEDEK, Z. *Int. Shipbuild. Progress*, **15** (1968), p. 78 (Mar.) [17 pp., 7 ref., 3 tab., 24 graphs, 6 diag.]

This is a publication of the Norwegian Ship Model Experiment Tank. The Author gives some simple expressions for thrust deduction and wake fractions, based on statistical analysis of the test results for some well-known geosim model families ("Victory," *Strinda*, *Meteor*). They permit simple extrapolation from model to ship.

26,865 **A New Idea in Propulsion.** THRING, M. W. *Ship and Boat*, **21** (1968), p. 16 (Oct.) [2 pp., 5 diag.]

The Author explains the working principles of marine air-cushion craft, and points out that some drawbacks of these craft could be overcome by the development of means for reducing skin friction without expending excessive air-fan power, together with means for using the water as the propulsion medium. One solution to this problem is the side-wall air-cushion vehicle, but this still needs substantial air-fan power; moreover, it requires a conventional landing-stage.

The side-wall A.C.V. nevertheless pointed the way for the Author's ideas for craft of improved power/speed characteristics. He suggests that the craft could be propelled by two endless belts fitted side-by-side under the hull. The belts could be provided with transverse paddle-blades; in this case each belt would consist of two narrow strips joined by the blades, and the space between the belts and the hull would be such that the flow was just laminar and the effective viscosity of the water therefore equal to its actual viscosity (instead of being 50 times greater as occurs when there is turbulence). An alternative to the paddle-belts is to have plain belts with air bearings between the belts and the hull; this system has the advantage of allowing the craft to move up a sloping beach or ramp for loading and unloading. With both systems, steering would be done by running the two belts at different speeds.

SHIP PERFORMANCE, STABILITY, AND MANOEUVRABILITY

(*See also Abstracts No.* 26,892 *and* 26,913)

26,866 **The Effect of Speed, Forebody Shape, and Weight Distribution on Ship Motions.** EWING, J. A. *Trans. R.I.N.A.*, **109** (1967), p. 337 (July) [8 pp., 18 ref., 4 tab., 41 graphs, 4 diag.; and Discussion: 1 p.]

This paper presents the results of computer calculations of the influences of various parameters on the motions of a ship in irregular head waves. The parameters concerned are: speed, forebody section shape, longitudinal radius of gyration, and longitudinal position of the centre of gravity (LCG). Heave, pitch, relative bow motion, and acceleration at bow and stern were computed for a Series 60 form of 0·70 C_b (extreme U sections forward), and three derivatives of it (moderate U, moderate V, extreme V); body plans are given. The wave spectra used as a basis were representative of Beaufort 5 and Beaufort 7 wave conditions in the North Atlantic; they were modified to take account of ship speed (F_n was 0·1, 0·15, or 0·2). The motion calculations were made using a program based on the theory of Korvin-Kroukovsky, with added mass and damping coefficient according to Grim. Statistically "significant" motion values were calculated for ship lengths of 100, 300, 600, and 900 ft. The response functions and "significant" values are presented graphically, the former on a base of wave-length/ship-length ratio and the latter on a base of ship length.

The principal finding is that V-shaped forebody sections give smaller "significant" motions than U sections at all ship lengths in the range considered. This seems to contradict the model-test results obtained by Swaan and Vossers for similar forms (see Abstract No. 17,534, July 1961). It is also found that ship speed has little influence on pitching in the F_n range studied, but considerable influence on heaving and on bow and stern accelerations, the effect being greatest for ships of about 500 ft.

Decreasing the radius of gyration reduces all the motion values except stern acceleration, which is slightly increased; for given ship length, a change of ± 2% in radius of gyration has a greater influence on pitching than any of the other variations studied. The influence of the LCG is complicated, and most apparent in heaving. In general, the present work confirmed the conclusions of an earlier paper based on Series 60 model data (see Abstract No. 24,756, Nov. 1966), *viz* that the ship characteristic which has most influence on pitching motion is length, and that heaving motion is influenced by ship speed and length.

26,867 **Shock Phenomena on a Trawler.** FERDINANDE, V. *Int. Shipbuild. Progress*, **15** (1968), p. 151 (May) [18 pp., 7 ref., 2 tab., 20 graphs, 2 diag., 2 phot.]

This is a theoretical investigation into shock phenomena experienced on the side trawler *Belgian Lady* during full-scale trials while sailing in rough head seas. The trials were arranged by the Centre Belge de Recherches Navales to study the performance and behaviour of trawlers; the motion results have already been published (see Abstract No. 22,817, Apr. 1965).

During the trials, accelerometers placed fore and aft recorded bad-weather shock phenomena as irregularities superimposed on the low-frequency oscillations. Slamming was indicated by an acceleration peak in the forward accelerometer followed by hull vibration. Due to the nature of the peak, and the fact that the forepart was apparently already in the water at the instant of shock, the Author was convinced that this shock phenomenon, though similar, was neither the "classical slamming" of the forebody on the water surface, or "destroyer slamming" caused by a build-up of hydrodynamic pressures on the flared bow.

The cause of these shocks on the *Belgian Lady* seemed to be the penetration of the forebody into the water. As the forebody sections have a V-form, an approach by strip theory can be based on the study of a triangular wedge penetrating the water surface. The basic technique used to tackle the problem was the Schwarz-Christoffel transformation, but in this case results were applied only to wedges with large deadrise angles. The piling-up of water was calculated and, as the transverse section of the fluid displaced by the wedge under the undisturbed water level is equal to the section of the wave rise above this level, this implied the determination of the wave rise profile. The pressure distribution on the side of the wedge was also calculated in conjunction with the hydrodynamic velocities and pressures in its vicinity.

The Author deals with the numerical calculation of hydrodynamic pressure and deck immersion of the triangular wedge section at some length, and presents the results graphically.

Finally, the results of the theory on the immersion of wedges was used to calculate the magnitude of acceleration peaks for the trawler sailing at a certain speed in a known sea state. In order to compare with distinctly measured values, a special trial run was chosen for the analysis. Characteristics of prevailing conditions, principal particulars of the trawler, and the known data used in the analysis are given. The results for the trawler, and a comparison of the measured and calculated magnitude of acceleration peaks at the bow, are presented graphically.

The Author concludes that for large vessels, where the proportion of

discontinuous hydrodynamic force variation to the mass is much smaller, these shocks are of less importance. For smaller vessels with a highly-flared bow, the same phenomenon might have serious effects. The present theory suggests moderation of this flare, provided that other seaworthiness qualities are not thereby affected.

26,868 **Note on the [I.M.C.O.] Corrections for Liquid Cargoes with a Free Surface** (in Italian). CARDO, A. *Tecnica Ital.*, **33** (1968), p. 17 (Jan.–Feb.) [4 pp., 2 tab., 1 graph]

26,869 **What's Her Range [of Stability]?** HATCH, G. N. *Ship and Boat*, **21** (1968), p. 34 (Apr.) [4 pp., 1 tab., 5 diag.]

The Author discusses the determination of the stability range—i.e. the limiting angle of heel beyond which the boat becomes unstable—of small privately-owned and commercial boats. He first explains in detail how the range can be calculated quite simply, within two or three days, by use of an integrator costing about £250 new or £100–£150 second-hand, illustrating his explanation by giving a worked example for a boat of 65 tons displacement. In making these calculations it is important to know which deckhouses can be considered watertight for short periods, because if the boat is capsized by a very heavy sea one complete revolution would normally take only about 15 seconds, during which very little water would be shipped through closed doors and windows. Correct trim to bring the longitudinal centre of buoyancy vertically below the centre of gravity is also of some importance.

The usually accepted criterion of " adequate stability " beyond an angle of heel of 10°–15°, which is taken to be covered by a GM between 3 and 4 ft, can result in great discomfort to the crew of the boat in high seas. A GM of 2 ft to 2 ft 6 in can be quite safe, and much more comfortable if the range of stability is large and the mass pendulum effect is large. The pendulum effect can be increased by moving heavy weights outward. Thus a twin-engined vessel with wing fuel tanks will have a larger pendulum effect than a single-engined hull with central double-bottom fuel and water tanks.

The stability range is affected by the geometry of the hull; high freeboard and large watertight deck structures help to maintain positive stability over a greater range of angles than low freeboard and large cockpits. Damping of roll by bilge keels is also beneficial; narrow vanes protruding some feet from the hull are much superior to long, shallow bilge keels. Adequate and rapid draining of open cockpits is also of importance.

26,870 **The Manœuvrability of Ships on a Straight Course.** HOOFT, J. P. *Int. Shipbuild. Progress*, **15** (1968), p. 44 (Feb.) [25 pp., 16 ref., 4 tab., 34 graphs, 7 diag.]

This is Report No. 99S of the Netherlands Ship Research Centre TNO. It describes an investigation of the manœuvrability, and the directional stability on a straight course, of 7 m (23 ft) models of a cargo-liner and a tanker; the tank width was 15·75 m (51·7 ft), and the water depth 1 m (3·28 ft). Details of the models (including body plans, end profiles, and sectional-area curves) and their propellers are given.

Kempf-type zigzag tests, and others in which the rudder angle was varied sinusoidally, were made with different rudder areas and ship speeds.

The test results are presented in a series of graphs. Values for the time constant of the ship (T) and the turning ability coefficient (K) in the equation of motion given by Nomoto (see Abstract No. 26,872 this issue) were derived by procedures which are explained, taking account of non-linear ship response.

The steering action of a human helmsman is described mathematically and related to ship response characteristics. As the coefficients of the response function for human steering of aircraft and cars are well known, by extrapolating to higher values for T and lower values for K response coefficients for large-ship steering can be found. The results of this extrapolation agree well with about 20 records from tankers of 50,000 to 100,000 tons d.w.

The interaction between ship, rudder, and helmsman when steering a straight course is discussed and a condition for course stability is derived. Finally, the steerability of two tanker forms at various speeds is discussed in relation to rudder area, hull form, and ship size.

It is concluded, *inter alia*, that a decrease in L/B ratio reduces the steerability, especially for the larger ships, and that a reduction also occurs even when all the principal dimensions are increased proportionally. Very large ships of low L/B (e.g. 300,000-ton tankers) will be difficult to steer, and no great improvement can be expected from an increase in rudder area.

In one of the appendices the Author considers the influence of rudder angle and steering on ship speed.

26,871 Experimental Studies on Ship Manœuvrability in Restricted Waters: Part I. FUJINO, M. *Int. Shipbuild. Progress*, **15** (1968), p. 279 (Aug.) [22 pp., 9 ref., 12 tab., 41 graphs, 4 diag., 2 phot.]

This is Part I of a two-part study; it deals separately with the influences of shallow water and of finite channel width.

The Author describes " planar-motion " tests, oblique towing tests, and tests in which rudder force and asymmetric hydrodynamic force were measured, all with systematic variation of water depth and channel width. This work was done at the Seakeeping Laboratory of Tokyo University. A planar-motion mechanism (forced yawing apparatus) was used to determine the coefficients of the equations of motion. This apparatus is briefly described with the aid of a diagram. The coefficients were determined by phase-analysis of the forces and moments necessary to make the model execute predetermined pure swaying and pure yawing motions. The apparatus prevented the model from surging, but allowed it to pitch, roll, and heave.

The two models tested represented fine and full hull forms. The fine form was a 0·59 C_b " Mariner " type, and the full form a model of the 0·80 C_b tanker *Tokyo Maru* (see Abstract No. 25,737, Sept. 1967). The principal particulars of both models are given; their lengths were 2·5 m and 2 m (8·2 ft and 6·56 ft) respectively. The forward speeds of the models were the scaled equivalents of 7 and 12 knots of the actual ships (7 knots only for the narrow-channel tests). The test conditions and results are set out and discussed in detail. Stability discriminants and indices (T′, K′, etc.) are given.

In the shallow-water tests the water-depth/draught ratios were 1·21, 1·5, 1·93, and 2·5 for the " Mariner " hull and 1·23, 1·5, 1·89, and 2·5

for the *Tokyo Maru*. It appears that shallow-water effects on added mass, added mass moment of inertia, and static (but not rotary) derivatives increase with speed of advance. Shallow-water effects become noticeable at water-depth/draught ratios between 2·5 and 2. As water depth decreases, the point of application of sway damping force moves far forward of amidships, and may eventually be forward of the corresponding point for yaw damping force, which makes course-keeping unstable; this occurs in the case of the *Tokyo Maru*, but not in that of the " Mariner". If water depth is decreased still further, straight-course stability returns but turning ability becomes worse. This dangerous depth range is, for the *Tokyo Maru*, between 1·6 and 2·6 times the draught, which means that, for manœuvrability tests, water depth should be at least four times the draught.

In the narrow-channel tests the effect of finite channel width on added mass was found to be greater on the fine than on the full hull form. The bottom widths of the trapezoidal (45°) narrow channel were 2 m (6·56 ft), 1·5 m (5 ft), and 1 m (3·2 ft), and the depth/draught ratios 1·2, 1·5, and 1·9 for both models. It was found that course-keeping on the centre line of the narrow channels used in the experiments was unstable; this condition seemed to be caused by very large asymmetric hydrodynamic force. The coefficients of the asymmetric hydrodynamic force and moment increase rapidly as water depth and channel width are decreased.

26,872 **Approximate Methods in Z-Steering Test Analysis.** KOELINK, J. T. H. *Int. Shipbuild. Progress*, **15** (1968), p. 35 (Feb.) [8 pp., 3 ref., 5 graphs, 7 diag.]

The Author first refers to a first-order differential equation evolved by Nomoto some years ago to describe the steering of ships. This expression, which gives the relationship between turning angular velocity and helm angle, contains two coefficients (K and T) which represent the steering qualities of the ship; these coefficients can be determined by the turning-circle test and the Z-steering (zigzag) test. (See also Abstract No. 25,571, Aug. 1967.)

The method of conducting a Z-steering test (which must be continued until a steady-state oscillation is obtained) is described with the help of a diagram. In practice an ideal Z-steering test cannot be achieved as rudder travel time would have to be zero, but a simple mathematical analysis of such an ideal test is presented which allows approximate methods to be checked against it. Two approximate methods (Norrbin's zero-crossing method and a describing-function technique) are explained and compared; the describing-function technique is shown to be the better of the two. A time-domain method is also given for that class of Z-steering test in which the describing-function method is not accurate enough. The accuracy of the describing-function and the time-domain methods has been verified by computer simulation; some illustrative data are given. It is concluded that, taking into account other sources of inaccuracy, the accuracy of the derived relations is adequate.

26,873 **Viscosity Effects on Lateral Forces and Moments during Ship Manœuvring** (in German). NIEMANN, U. *Schiff u. Hafen*, **19** (1967), p. 891 (Dec.) [3 pp., 4 graphs, 2 diag.]

This paper was read at the Nov. 1967 Meeting of the Schiffbautechnische

Gesellschaft, and is followed by a brief account of the subsequent discussion.

The paper gives a short account of a theoretical and model study on the lateral forces and horizontal moments affecting ships during turning, with particular reference to viscosity effects.

26,874 **Wind Forces on Upper Works [including Superstructures, Masts, etc.]** (in German). WAGNER, B. *Schiff u. Hafen*, **19** (1967), p. 894 (Dec.) [7 pp., 8 ref., 18 tab., 11 graphs, 5 diag.]

This paper was read at the Nov. 1967 Meeting of the Schiffbautechnische Gesellschaft, and is followed by a brief account of the subsequent discussion.

The Author gives an account and extensive results of tests in which wind forces and moments on above-waterline models of various types of ship were measured. The models included variations (in respect of superstructure position and of load condition) of the same ship type, and also included a hydrofoil craft. It is pointed out that wind pressure on the decks of a rolling ship can be important.

STRUCTURAL DESIGN AND ITS APPLICATIONS

26,875 **Transverse Strength of Tankers—Finite Element Applications.** RØREN, E. M. Q. *Europ. Shipbuild.*, **17** (1968), p. 42 (No. 3) and p. 58 (No. 4) [16 pp., 15 ref., 29 diag.]

Much of the recent work on the transverse strength of tankers has been based on conventional methods of the structural analysis of framed systems by the application of beam theory in some form. The various members of the structure are assumed to have beam properties with the members joined at certain more or less arbitrarily chosen joints or nodes; and internal stress resultants are found by some analytical method which may be of the flexibility or stiffness type. Structural-analysis programs are available for using this method in a computer. The Author illustrates the application of the method to a typical tanker web frame, using slope-deflection relations.

One of the chief disadvantages of this method is that many frames resemble plates with large cut-outs rather than the traditional more or less slender frames. The corner regions have very considerable effects; " beam " regions are difficult to define, and may, in fact, scarcely exist. Another difficulty is presented by the choice of the " model " of the frame on which the analysis is to be carried out.

These and other disadvantages are avoided by using the finite-element method, in which the complete structure is resolved into a suitable mesh or set of small elements, each of which is regarded as a plate subjected to a plane-stress condition. The technique can be applied to structures of almost any configuration; the final accuracy depends primarily upon the fineness of the mesh and its layout in regions near curved boundaries to give elements of suitable shape. For the efficient use of this analysis, data-generating routines are required to provide data needed before the analysis can be started on the computer. Such routines have established the necessary data for two typical transverse frames of tankers chosen by

the Author to illustrate the application of the method, after he has explained its basic principles.

The mesh chosen for the first frame resolves it into 343 plate elements, 188 bar elements, and 406 nodes. Characteristic loading cases have already been programmed for the computer in detail, and the user has merely to specify his desired condition through a simple code. The loading condition chosen by way of example is light draught, empty side tank, full centre tank. Diagrams show the calculated stress distribution in the wing-tank transverse, in the bottom transverse of the centre tank, and in the lower T-joint; and also certain shear forces and moments. In some cases the results are compared with those calculated by the conventional method.

The second frame is also analysed by both methods. For the initial calculations by the finite-element method a rather coarse " macro- " mesh is used, but to determine the stresses near a corner, and the effect of varying the shape of the flange there, a finer "micro-" mesh is applied to that portion of the frame, comprising about 410 elements and 340 nodes. Two loading conditions are studied for this frame.

The method can be applied to the complete hull, including the external shell as well as bulkheads and internal primary members. The Author shows a proposed " module " for this purpose which comprises a total of 50 nodes inter-connecting a three-dimensional arrangement of membranes and bars. He expects that future developments will include the solution of problems relating to the vibration and stability of plates and panels.

26,876 **Stress Concentration in Way of Hatch Corners—Effect of Changes in Corner Radii and Spacing of Openings.** BELL, A. O., and RICHARDSON, W. W. S. *Shipp. World & Shipb.*, **161** (1968), p. 567 (Mar.) [4 pp., 2 ref., 2 tab., 5 graphs, 11 diag.]

This is the third and final report on an investigation carried out by the British Ship Research Association. The earlier reports concerned experiments to determine the effectiveness of corner insert plates and parabolic-shaped corners in reducing the stress concentrations at hatch corners (see Abstract No. 25,883, Nov. 1967).

The present part deals with investigations into the effect of altering the corner radii and the spacing between openings. The measurements were made on the specimen used for the comparative tests on radiused and parabolic corners, the corner radii being altered after each series of tests.

The test specimen is a plate 0·25 in thick, 24 ft long, and 10 ft wide, with four 37·5-in square hatch openings. There are three transverse stiffeners, and the plate edges are stiffened by flat bars and supported at intervals by pillars. Diagrams illustrate structural details of the specimen. Corner radii from 4 to 10% of the breadth of the opening were investigated. The distances between the openings are varied from 0·1 to 1·5 times the breadth of the openings.

The method of preparing the corner radii used for the earlier tests was again employed. Electric-resistance strain gauges having a base length of 0·125 in were used to measure the circumferential stress at the hatch corners. Five gauges were attached at each corner. Similar gauges having a 0·5 inch base length were placed at six transverse sections abreast the openings; these were arranged in pairs on either side of the plate to measure " heart-of-plate " strains. A load of 80 tons in axial tension

was regarded as datum, and was applied for all tests on the specimen. Its purpose was to remove initial unfairness. This load was increased up to 150 tons by increments of 10 tons.

Stress distributions at the six gauged sections abreast the openings are shown. These measurements are similar to those obtained in the earlier experiments, the stress at the edges of the opening being greater than at the outside edges of the plate, and the difference increasing with the ratio of inter-hatch spacing to breadth of hatch. The location of peak stress appeared to be mainly governed by this ratio. Stress-concentration factors were obtained by dividing the peak stress at the corner by the mean applied stress for the cross-section abreast the openings. These stress-concentration factors are tabulated and plotted against the ratios of corner radius and inter-hatch spacing to hatch breadth.

The main objective of the whole investigation has been to obtain data so that basic stress-concentration factors for a rectangular opening in an infinite plate may be modified to take account of the geometrical and structural features of the particular design.

26,877 **[Statistical] Principles Governing the Distribution of Permanent Set in Deck and Shell Plating** (in Russian). BRIKER, A. S. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 82 (1967), p. 19 [7 pp., 3 ref., 3 tab., 2 graphs]

In this paper the Author presents and analyses statistical data concerning the magnitudes of permanent sets in the deck and bottom plating of vessels which have been in service for a long time. He deals only with vessels having transverse framing, because the general and local strength of such vessels is particularly affected by such distortion, especially when it is wavy. It is shown that the magnitude of permanent set in each sub-panel between stiffeners can be treated as a random variable, and that distribution curves of this variable (based on frequency of occurrence on any given hull) approximate quite well to a Pearson type III distribution, which is unimodal and has unlimited range in one direction. A relation is established between stiffener spacings and the permanent set in each sub-panel of the plating; this is illustrated by a graph.

26,878 **Investigation of the Principles [Governing the Static and Dynamic Pressure] of Grain Cargoes, using a Test Rig Simulating Rolling, Pitching, [Wave Impact, and Vibration]** (in Russian). TUKHKANEN, O. I. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, **88** (1967), p. 97 [5 pp., 3 tab.]

The main purpose of the investigation was to provide a basis for the strength design of grain bulkheads, shifting boards, etc.

26,879 **Prediction of the Low-Cycle Fatigue Life of Pressure Vessels.** PICKETT, A. G., and GREGORY, S. C. *A.S.M.E., Paper No. 67 Met 3, presented* 3-5 Apr. 1967 [11 pp., 4 ref., 14 graphs, 1 diag., 1 phot.]

The bases for A.S.M.E. Boiler and Pressure Vessel Code (Section III) fatigue evaluation procedures, the fracture-mechanics approach to fatigue life analysis, and Stowell's "notch stress analysis method" are reviewed. Fatigue-life predictions are compared with the results of tests on materials, models, and full-size pressure vessels. A proposed design procedure, using the "notch stress analysis" method and experimental results, is

presented. It is based on the likelihood that a crack-like flaw or defect will exist in a highly-strained region very early in the low-cycle fatigue life of initially sound specimens, and that this life can be analysed as crack-propagation life from such a crack. *Inter alia*, this proposition provides a cumulative-damage law which is more logical than Miner's rule and which tests have shown to be valid.

WELDING AND OTHER METHODS OF CONSTRUCTION

26,880 **Welding of Steel Coated with Priming Paint. Recommended Standard Welding and Cutting Tests for the Assessment of the Toxicity of Paint Primer Thermal Decomposition Products. Revised Edition—1968.** *Brit. Weld. J.*, **15** (1968), p. 468 (Sept.) [2 pp.]

The first (1964) edition is covered by Abstract No. 22,957 (May 1965). It is hoped that the present revised edition will remove some anomalies relating to the period of validity of the test reports.

Procedures are described for preparation of materials, preliminary tests to determine the nature of the toxic constituents of the fume, flame-cutting tests, and manual metal-arc welding tests. Instructions concerning the analysis of samples, the presentation of results, and the validity of test reports are also given.

26,881 **Toxic Constituents of Welding Fumes.** STEEL, J., and SANDERSON, J. T. Reprint from *Annals of Occupational Hygiene*, **9** (1966), p. 103 [9 pp., 6 ref., 5 tab.]

The paper describes experiments designed to determine the amounts of certain toxic metals present as impurities or additions in electrode coatings, and to assess the potential environmental hazard during welding operations.

Twelve covered electrodes in common use for the welding of mild steel were tested in laboratory and simulated field conditions. The results show that existing methods for assessing welding-fume hazards, based on iron-oxide or total fume concentration, can be dangerously misleading. All twelve coatings proved to be hazardous when assessed by a " synergistic hazard index " (S.H.I.).

Attention is also drawn to some common faults in ventilation practice.

26,882 **[Experiments on] the Use of CO_2 for [Semi-Automatic] Underwater Welding [with Thin Wire].** MADATOV, N. M. *Welding Production*, **14** (1967), p. 21 (Dec.) [5 pp., 6 ref., 2 tab., 3 graphs, 8 phot.]

Although CO_2 has a beneficial effect on metal transfer in underwater welding, the drawbacks make its use inadvisable, except possibly for downhand welding in shallow water.

26,883 **Glued/Welded Joints in Shipbuilding [Made by a Combination of Spot-Welding and Adhesive Bonding]** (in Russian). BOCHKAREV, V. P., DOLGORUKOV, V. V., and GLEVITSKAYA, T. I. *Sudostroenie*, No. 9 (1968), p. 67 (Sept.) [2 pp., 4 diag., 1 phot.]

A method is described of joining aluminium structural components of high-speed vessels by a combination of spot welding and adhesive bonding

using a special synthetic adhesive, the composition of which is given. The advantages of this type of joint are: greater damping of vibration, more uniform distribution of applied loads, lowering of stress concentration at the welded spots, and increased fatigue strength.

SHIPBUILDING (GENERAL)

26,884 **Technical Progress—Today and Tomorrow.** ABRAHAMSEN, E. *Norwegian Shipp. News*, **24** (1968), p. 766 (6 Sept.) [4 pp., 3 graphs]

This is a paper presented at the Second International Shipping Exhibition, Oslo, 20–29 May 1968.

The Author, who is the managing director of Det Norske Veritas, discusses the problems involved in the design and construction of very large tankers. He considers it quite feasible to apply advanced technology in this field to ensure that these ships possess a greater degree of safety, and are less hazardous to operate, than smaller ships.

Two years ago Det Norske Veritas completed a technical design study for a 500,000-ton d.w. tanker and, as a result, issued certain recommendations regarding the structural details and scantlings of such a ship (see Abstracts No. 25,676, Sept. 1967, and 25,662, Sept. 1968). Further studies have now been completed for tankers up to 1·3 million tons displacement; by systematically varying the parameters of length, beam, depth, and draught, and considering several alternative structural arrangements, it has been shown that, technically, there are no obstacles in constructing ships of this size, or even larger. As a result, the optimum size of a tanker for a given service will be governed mainly by the state of the market and the availability of suitable loading and discharging facilities. However, some operational flexibility may have to be sacrificed to obtain the lowest ton/mile costs.

The events which have affected the demand, supply, and movement of crude oil since the end of the Second World War are discussed. Particular reference is made to the closure of the Suez Canal and to the increased demand from Europe and Japan for oil from the Persian Gulf, all factors which have led to the building of large crude-oil ships and the gradual phasing out of smaller ones carrying mixed grades. If this trend (which is shown graphically) continues, it is possible that by 1970 half the world's tanker tonnage will consist of ships exceeding 60,000 tons d.w.

The investment required for a ship per unit carrying capacity is shown against deadweight capacity multiplied by the ship speed; it decreases considerably as ship size increases. The biggest decrease in price per deadweight ton occurs when increasing the ship size from 20,000 to 100,000 tons d.w., but even beyond this size considerable reductions may be obtained, as shown in a graph. However, the further reductions obtainable by increasing deadweight beyond about 400,000 tons may be small.

The reduction in the investment required for ships up to 400,000 tons is mainly due to the fact that the steel weight needed per ton of cargo capacity is reduced up to this point. Beyond this, however, the weight of the transverse material increases rapidly in relation to the total steel weight, although that of the longitudinal material (which constitutes the greater part of the total steel weight in a large tanker and is mainly

determined by the wave bending moment) remains remarkably constant and independent of ship size beyond 150,000 tons.

The most economical method of increasing the cargo-carrying capacity of a ship in the design stage is to increase only the breadth and depth, as length is the most costly parameter to alter. Further economies may be achieved by the use of certain high-strength steels, but care must be exercised in the structural design to ensure that they are correctly used, and that their characteristics are taken into account, especially during welding and prefabrication.

Regarding the powering of very large tankers, the Author describes, with the help of a graph, how the power needed to attain a given economical service speed of about 16 knots increases as displacement to 2/3 power, resulting in a relative reduction in power requirement and fuel consumption as ship size increases. Since these reductions are larger than the relative reductions in investments, the most economical speed will tend to increase with ship size, although this may not be desirable for ships operating on fixed low time-charter rates.

In tankers exceeding 300,000 tons a higher propulsion efficiency can be obtained with a twin-screw installation; this improves reliability, assists manœuvring, and attracts more favourable insurance rates. Because of the heavy loading on propellers of large tankers, propulsive efficiency generally decreases as ship size increases; this can be alleviated to a certain extent by reducing the propeller revolutions to about 80 r.p.m. Kort nozzles or controllable-pitch propellers, used in conjunction with fully-automated steam-turbine or medium-speed Diesel propulsion units running on heavy fuel, may well prove very suitable for large tankers with respect to powering, manœuvring, and stopping.

The main types of casualties in the tanker trade are discussed, and the importance of accurately assessing the potential hazards to which very large tankers are exposed is emphasised. Crews must be well trained in the use of modern navigational aids and sophisticated automation systems. Particular attention must be paid to safety during gas-freeing, loading, and discharging, as increase in ship size usually leads to higher gas concentrations per unit deck area. Monitoring of the gas concentration at critical points, and reduced loading rates, may be necessary under adverse weather conditions. A very high standard of fire-fighting and damage control must be attained by all the crew.

See also Abstract No. 26,398 (May 1968).

26,885 **Sea-Going Vehicular Ferries (Some Aspects of Standardisation).** BARCLAY, C. *Paper read at Europort Congress*, 1965 [9 pp., 1 tab., 5 diag.]

The Author reviews the distinctive features of three kinds of vehicle ferry, viz., short-haul (up to 100 miles), medium-haul (up to 400 or 500 miles), and long-distance.

He then discusses the possibility of standardising certain features of stern-ramp ferries and terminal link-span installations, so as to give a much wider choice of terminals. The connection between ship and link-span is the feature of ship-shore connection which lends itself most easily to standardisation. A table is included comparing the relevant characteristics of eight ferries now in service.

Standardisation is particularly advantageous for short-haul and medium-haul ferries loading and unloading in tidal waters, but its

application to long-distance ferries also would enable them to use the same facilities; they could then run on the shorter routes during " slack " periods when they might otherwise be laid off. Standardisation should ensure a more efficient handling operation, a reduction in turn-round time, and a consequent reduction in costs.

The paper suggests that a committee should be appointed to examine standardisation, and to make recommendations on the following points:—

1. Definition of a minimum distance between quay and the link span, to allow for berthing vessels of greater beam. It is suggested that the centre line of the link span should be parallel to, but not less than 14 m (45·9 ft) from, the quay.
2. Agreement on a standard design of stern door, preferably of the downward opening hinged-ramp type, and on a minimum door width to suit vehicles up to 3 m (9·8 ft) wide.
3. The proposal that the height of the car deck aft, or the height of the sill aft, above the waterline should be kept between limits of 1·3 m and 3 m (4 ft and 9·8 ft).
4. The proposals that the ship's fender height should exceed 1·2 m (4 ft) above the waterline, and that the maximum distance between the aft sill and the upper lip of the ship's fender should be fixed, e.g. at 0·5 m (1·6 ft).
5. Formation of a ferry operators' association to advise owners on matters relating to ferry ships.

26,886 ***Esso Mercia.*** **New Esso Tanker has MST-14 Steam Turbine.** *Shipp. World & Shipb.*, **161** (1968), p. 560 (Mar.) [8 pp., 2 ref., 4 tab., 14 diag., 6 phot.]

The *Esso Mercia* is a large all-aft tanker, with a raised forecastle, a prominent projecting bulbous bow, and an AG Weser stern bulb; she was built for the Esso Petroleum Co. by AG Weser of Bremen, and is the largest tanker yet built in their yard. She is classified by the American Bureau of Shipping as ✠A1 (E) " Oil Carrier " ✠AMS. Her principal particulars are:—

Length, o.a.	1,010·2 ft (307·90 m)
b.p.	950 ft (289·55 m)
Breadth, moulded	146 ft (44·40 m)
Depth, to upper deck	76·4 ft (23·30 m)
Draught, summer	59 ft (17·98 m)
Deadweight, corresponding	166,820 tons
Displacement, summer	194,083 tons
Service speed	16 knots

The hull is all-welded, and has a rounded gunwale plate. The cargo section is subdivided into five centre and ten wing tanks, sets Nos 1 and 5 being about twice as long as the others; the after ends of No. 5 wing tanks are partitioned off to serve as sludge tanks. The longitudinal bulkheads extend through the machinery space, serving as boundaries for wing bunker tanks.

No. 2 wing tanks are used for water ballast only. Ballast is also carried in the fore and after peak tanks, in deep tanks forward and abaft the cargo section, and in special wing tanks aft of the wing fuel bunkers. Nos 3

and 4 tank sets have been designed to act as flume-type roll stabilisers. Over-the-top ballasting of selected cargo tanks which have been previously emptied is possible while oil cargo is being discharged.

The cargo-pumping system incorporates a centreline duct which is an integral part of the ship's structure. The independent cargo-stripping system is based on a conventional pipework layout, and is illustrated by a diagram. Oil cargo is handled by four AG Weser turbine-driven Eureka centrifugal pumps, each capable of discharging 3,000 tons/hr (sea water). There is also a turbine-driven ballast pump of the same capacity. The cargo stripping pump is of 365 tons/hr capacity.

All cargo-handling operations can be carried out from a central control station located below the bridge, provided with remote control of all main cargo gate valves and remote indication of the ullage and temperature of each cargo tank. The gauging equipment has been provided by Dobbie McInnes Ltd. Hydraulic actuating motors, with mechanical connection to the valves, are located on deck for ease of maintenance.

Deck mooring machinery is hydraulically operated, the power being supplied by three 300-h.p. electrically driven pumps located on the peak deck aft, and by a 900-b.h.p. Paxman Ventura Diesel engine driving a hydraulic pump located under the forecastle deck.

Among the equipment provided by the International Marine Radio Co. is a 100-watt reserve transmitter, with "Mayday" RT facilities on 2,182 kc/s, which is claimed to be a unique feature.

Safety equipment includes a foam mains system supplying a number of dispersing monitors located at intervals along the deck, and a remotely operated emergency fire pump, powered by the Diesel-hydraulic unit forward.

There is air-conditioned accommodation for an "integrated" crew of 33, all in single cabins. Marinite veneered with laminated plastics was used for bulkheads, linings, and ceilings.

The main engine is a non-reheat version of the MST-14 package-type steam-turbine set, built jointly by AG Weser and the designers (U.S. General Electric) who supplied the rotating parts. (See also Abstract No. 25,512, July 1967.) It is a two-casing cross-compound design with an axial-exhaust L.P. turbine, and has a rated output of 30,000 s.h.p. at 80 r.p.m. through dual torque-path reduction gearing. Inlet steam conditions are 850 lb/sq in, 950° F. The main machinery can be controlled from the bridge as well as from the machinery control room by a Siemens remote-control system. Siemens also provided main-machinery instrumentation, including a 108-point data logger, and an efficiency computer.

The Alcunic propeller has a diameter of 30 ft (9·2 m) and a pitch of 19·5 ft (6·64 m); it weighs about 52·5 tons.

The single main boiler is of a new design developed by Babcock & Wilcox in the U.K., and was built by their German associates in Oberhausen. This "Bi-Drum" boiler is the first of its type to be installed on board ship. (See Abstract No. 26,905, this issue.)

The ship's A.C. supply is obtained from an AG Weser/Hansa turbo-alternator set of 1,000 kW output. The standby set has a 12-cylinder turbocharged Paxman Ventura Diesel engine (1,200 r.p.m.) coupled to a Hansa 750-kW alternator. The emergency set has a six-cylinder Paxman Diesel driving a Hansa alternator of 200 kW output.

A deadweight scale, tank-capacity tables, general-arrangement drawings, a midship section, steel plans, and structural drawings are included in the article, together with a photograph illustrating the use of scale models to find the optimum machinery arrangement in the engine room.

26,887 **Japanese Full Container Ship Construction.** *Shipbuild. Shipp. Rec.*, **111** (1968), p. 761 (31 May) [3 pp., 1 tab., 19 diag.]

General-arrangement drawings and tabulated principal particulars are given for several ships, designed exclusively for the carriage of containers and due for completion in the period Aug.–Oct. 1968. The builders and owners are: Hitachi, for Yamashita-Shinnihon; Ishikawajima-Harima, for Japan Line; Kawasaki, for Kawasaki Kisen; Mitsubishi, for Nippon Yusen Kaisha (two sister ships) and Mitsui O.S.K. Lines (two sister ships). All these vessels are of similar, though not identical, dimensions and layout. Length b.p. is the same for all (175 m, i.e. 574·1 ft); breadth ranges from 25 to 26 m (82 to 85·3 ft), depth from 15·3 to 15·5 m (50·2 to 50·9 ft), and full-load draught from 8·95 to 9·5 m (29·4 to 31 ft). Block coefficients are in the range 0·56–0·58, and service speeds between 22·25 and 22·6 knots. In all cases the machinery (single low-speed Diesel) and superstructure are three-quarters aft, the forefoot is bulbous (though the size of the bulb varies), and the stern frame is of clearwater type with horn rudder. The Kawasaki design, unlike the others, has a raised quarterdeck and a transom stern; all designs have a forecastle, and wing tanks in way of the container holds. The container capacity is 1,400–1,430 standard 20-ft containers (six tiers in the largest holds and two tiers on hatch covers). Provision is made for refrigerated containers, but the number varies. There is no handling gear.

26,888 ***Atrevida*—East Asiatic Co.'s 10,880/14,200-ton 21½-knot Cargo Liner.** *Motor Ship*, **49** (1968), p. 61 (May) [5 pp., 2 ref., 7 diag., 4 phot.]

The *Atrevida* was built for the (Danish) East Asiatic Co. by Nakskov Shipyard Ltd, Nakskov, Denmark. The ship is an open/closed shelter-decker, and conforms to Lloyd's highest classification. She has a deadweight capacity increased by about 1,000 tons over previous East Asiatic ships, and a slightly higher service speed. Her principal particulars are:—

Length, o.a.	167·06 m (548 ft)
b.p.	155 m (508·5 ft)
Breadth, moulded	24·76 m (81·25 ft)
Depth, to upper deck	13·3 m (43·6 ft)
to second deck	10·12 m (33·2 ft)
to third deck	7·19 m (23·6 ft)
Draught, open	8·55 m (28·1 ft)
Corresponding deadweight	10,880 tons
Draught, closed	9·9 m (32·5 ft)
Corresponding deadweight	14,200 tons
Gross register	8,930 tons
Net register	5,129 tons
Hold capacity, including deep tanks	721,000 cu ft (grain)
Refrigerated capacity	84,000 cu ft
Service speed	21–22 knots

The ship is designed basically for general and refrigerated cargoes, vegetable oils and latex, but provision has also been made for containers. There are four continuous decks, a raised forecastle, and a fairly long poop deck carrying the superstructure. A spade rudder is fitted below the transom stern. There are six holds in all; five are forward of the engine room (No. 1 being below the forecastle) and one (No. 6) aft. Nos 1 and 6 holds can be refrigerated down to $-25°$ C ($-13°$ F), and are divided into four and seven compartments respectively. There are heated deep tanks (with stainless-steel coils) for liquid cargoes at the base of the watertight bulkheads separating Nos 2, 3, 4, and 5 holds, and two unheated wing tanks in No. 2 upper tweendeck. All holds are mechanically ventilated and the recirculation system (with thermostatically-controlled steam heaters) is remotely controlled from the chartroom.

There are large twin athwartship hatches in all decks to Nos 3, 4, and 5 holds. A total of 354 standard $20 \times 8 \times 8$ ft containers can be carried, stacked in the holds in five tiers of nine in way of each hatch, and on the weather deck in two tiers on top of the strengthened hatch covers; the tank top in way of the container stowage space in the holds is also suitably stiffened. Wing tanks in Nos 3, 4, and 5 upper tweendecks provide additional strength and can be used for ballast water.

MacGregor single-pull hatch covers with automatic raising and cleating are fitted on the weather deck, and hydraulically-operated flush-fitting covers in the tweendecks, where fork-lift trucks can operate; the cover controls are on the weather deck. Nos 1 and 6 holds are each served by two 5-ton derricks, and provision has been made for a 5-ton crane to be fitted forward of No. 1 if required. No. 2 hold is served by two 5-ton derricks mounted on the forward bipod mast, and also by a 5-ton ASEA rail-mounted deck crane travelling longitudinally over No. 3 port-side hatch. No. 3 hold is served at its after end by two 5-ton and one 25/60-ton derricks (all mounted on the after bipod mast), and also by the 5-ton crane already mentioned. Nos 4 and 5 holds are served by two 5-ton cranes (one port and one starboard) which travel the full length of both hatches; these two holds are also served by two 5-ton and one 25/60-ton derricks mounted on the after bipod mast, and by two 10-ton and one 25-ton derricks mounted on the fore side of the superstructure. The 5- and 10-ton derricks have Thrige-Titan electric topping-lift and preventer-guy winches. The heavy-lift derrick has a guyless self-slewing rig designed by the shipbuilder. All derrick consoles are enclosed in small cabins.

The automation and remote-control systems are described. They are basically the same as those fitted in the *Arosia* (see Abstract No. 25,596, Aug. 1967) and the earlier *Ancona* and *Andorra* (see Abstract No. 24,655, Sept. 1966), but include some minor modifications. In a total crew of 36 there are 11 engine-room personnel (one more than in the *Arosia*) comprising four senior engineers, three junior engineers, one electrician, and three motor men; the additional senior engineer is required for the larger refrigeration plant. The engine room is periodically unmanned.

The main engine is a 12-cylinder B. & W. type 1274-VT2BF-160 with a maximum continuous rating of 19,800 b.h.p. at 119 r.p.m. It can operate on fuel of up to 3,500 sec. Redwood No. 1, although 1,500-sec. fuel will normally be used. The engine drives a KaMeWa controllable-pitch propeller. A comprehensive alarm system monitors operation of

the main machinery, the system being interlocked with the starting/fuel-control lever. There is a combined control system for the main engine and the KaMeWa propeller, incorporating a Woodward PG governor on the engine and a similar governor for the hydraulic actuating gear of the propeller. Engine output and propeller pitch are controlled from a single lever on the wheelhouse console; the arrangements are duplicated in the machinery control room (which is on the port-side upper platform). The control lever actuates a cam-operated air telemotor system connected to the governors already mentioned. The operation of the telemotor system and associated protective devices is described. Automatic shut-down of the engine in the event of lubricating oil-pressure failure is effected by breaking the circuit to the shut-down solenoid on the Woodward governor through a 30-sec. time delay to allow for starting of the stand-by pump.

Electrical power is supplied by three 970-kVA Thrige-Titan Autovolt alternators, each driven by a seven-cylinder B. & W. 726-MTBH-40 Diesel engine. These sets are disposed in a longitudinal row to port of the main engine. Automatic supervision of the electrical machinery is by pressurestats and thermostats on the lubricating and fresh-water systems, connected over a time relay to the solenoid shut-down on the Woodward governors. Automatic starting of the generators is not provided.

The automatic control systems for the main-engine services and the fuel separators are similar to those in the *Arosia.*

The Freon 22 fully-automatic refrigeration plant is of the direct-expansion type. Five Sabroe eight-cylinder SMC8-65 type compressors, located in the after end of the forecastle, serve No. 1 hold, and a similar set of five serve No. 6 hold. A SMC4-65 compressor in the engine room provides for domestic requirements.

There is a 30 cu m (1,060 cu ft)/hr Diesel-driven fire pump in the forward refrigeration-compressor compartment. Two 90 cu m (3,178 cu ft)/hr electrically-driven fire pumps in the engine room can be combined with a third pump of 150 cu m (5,297 cu ft)/hr for operating the ballast system. Two 100 cu m (3,531 cu ft)/hr ballast ejectors are fitted in the pump room forward.

There is a CO_2 total-flooding system for the engine room and holds, and a Ellehamer foam smothering system for the machinery spaces. Fire detection is by Cerberus ion detectors.

A KaMeWa bow-thrust unit, driven by a Thrige-Titan squirrel-cage motor, is fitted forward of No. 1 hold. Steering gear is of the Hastie electro-hydraulic four-ram type and has an electric control system designed by the shipbuilders; there is a separate steering wheel and control system for each of the two pump motors.

There are general-arrangement and machinery-layout drawings, and photographs of the control room, engine room, and cargo-handling gear.

25,889 ***Maihar*** **New Brocklebank Ship from Swedish Yard.** *Shipp. World & Shipb.*, **162** (1968), p. 1051 (July) [10 pp., 2 tab., 1 graph, 10 diag., 8 phot.]. See also *Shipbuild. Shipp. Rec.*, **111** (1968), p. 753 (31 May) [5 pp., 1 tab., 1 graph, 4 diag., 3 phot.]

The cargo liner *Maihar* and her sister ship *Mahsud* have been built by Lindholmens Varv A.B. Gothenburg, for Thos. & Jno. Brocklebank Ltd.

of Liverpool; they will operate in the new Cunard-Brocklebank service from the Gulf of Mexico to Ceylon and the Bay of Bengal. The original design, based on extensive studies by the owners and builders in conjunction with the Yarrow-Admiralty Research Department, was intended to run between the U.K. (especially Manchester) and India/Pakistan (especially Calcutta) via the Suez Canal; the exigencies of this route dictated some of the main dimensions. The *Shipbuild. Shipp. Rec.* article states that the *Maihar* is the first "all-metric" vessel to sail under the British flag. Her principal particulars are:—

Length, o.a.	153·8 m (504·6 ft)
b.p.	143·25 m (470 ft)
Breadth, moulded	19·2 m (63 ft)
Depth, moulded, to upper deck	11·43 m (37·5 ft)
to second deck	7·77 m (25·5 ft)
Draught, summer	8·53 m (28 ft)
Deadweight	11,846 tonnes
Displacement	17,025 tonnes
Lightweight	5,184 tonnes
Block coefficient	0·699
Cargo capacity:	
grain	16,217 cu m (572,700 cu ft)
bale	17,307 cu m (611,910 cu ft)
refrigerated	566·3 cu m (20,000 cu ft)
liquid (edible oil)	500 tons
Service speed	17·5 knots
Classification	Lloyd's Register ✠100 A1 ✠LMC

The hull has a straight raked stem (a bulbous bow was tank-tested but found to be of little use), a transom stern, and a clearwater stern frame with horn rudder. There are five main holds, the engine room and superstructure being located between Nos 4 and 5; a poop deck extends from the superstructure to the stern. The transverse bulkheads separating Nos 2, 3, and 4 are vertically corrugated. The second deck (tweendeck) is stepped up to No. 1, and the upper deck has strong sheer from the after end of No. 2. There is a short forecastle. The lower part of No. 1 hold is subdivided into deep tanks (provided with heating coils and pumping trunks) which can be used for general cargo, edible oil, or water ballast; cars can be stowed on top of these tanks, below the second deck. No. 2 hold has a lower tweendeck. Nos 3 and 4 are pillarless, but have partial centreline bulkheads. The lower wings of No. 5 (in way of the shaft tunnel) are used for fresh-water and other tanks. The machinery casing is relatively small, leaving a considerable second-deck area above the engine room; this has been subdivided into seven refrigerated lockers grouped around an access space reached through a flush hatch (2·8 × 5·6 m, i.e. 9·2 × 18·4 ft) in the upper deck above. This hatch is in the floor of a transverse passage through the superstructure, and is served by a Nordströms Linbanor retractable side transporter, which can operate with a 2·5-ton load against a 5° list.

Holds Nos 3 and 4 have large triple hatches, the centreline openings (14·7 × 6·4 m, i.e. 48·2 × 21 ft) being twice as wide as the side ones.

Nos 1, 2, and 5 have single hatches. The weather-deck hatch covers are of single-pull type and the others (flush) of Mini-pack type; all were supplied by Swedish MacGregor. About 86 20-ft containers can be carried on the weather-deck covers (which are suitably strengthened), and another 48 below deck. There is at least 11 ft clear headroom on the second deck at Nos 1, 2, and 3. Masts, etc., had to be kept low (less than 71·5 ft above light waterline) to permit passage through the Manchester Ship Canal. The cargo-handling gear includes five 5-ton deck cranes (two between Nos 1 and 2, two between Nos 3 and 4, and one just forward of No. 5); these are of a new ASEA design in which all motions are powered by simple squirrel-cage motors with thyristor control circuits. A mast with crosstrees, between Nos 2 and 3, carries a 60-ton and two 5-ton derricks for No. 3 and a 10-ton derrick for No. 2; there are also a 25-ton derrick at the superstructure front, and a 10-ton derrick on a post with crosstrees aft of No. 5. The derricks are of Hallén slewing type, and are served by winches which (like the windlass and the three 8-ton mooring capstans) have Siemens pole-changing A.C. drives.

Each hold is ventilated (mechanical supply, natural exhaust) by a reversible fan rated at 5,000 cu ft/min per 100,000 cu ft (grain) served. There is special ventilation for hazardous cargoes in No. 1 tweendeck and orlop deck. The refrigerated lockers can be held at temperatures down to -20° F by a Stal R22 plant on the upper engine-room flat; this plant also serves the three main air-conditioning systems, which were supplied by Svenska Fläktfabriken and are as follows: (i) A Duovent dual-duct cabin-control system for the European accommodation (11 persons). (ii) A Regovent group-control system for the Asian accommodation (44 persons, in poop). (iii) A separate system for the workshop and machinery control room (both on starboard side of upper engine-room flat). *Shipbuild. Shipp. Rec.* gives the ratings of these systems. Walter Kidde smoke detectors monitor all cargo compartments, and there is a Minerva fire detection and alarm system for the engine room. The engine room has a Kidde CO_2 total-flood system; the cargo spaces are protected by a Kidde-Lucas oil-burning inert-gas generator (44,000 cu ft/hr) located in the emergency-generator room in the main mast house. A cargo space can be flooded with inert gas without stopping to close it up, and can be kept in this condition for long periods. Bulkheads and ceilings in the accommodation are of Marinite.

A N.P.L. passive-tank stabilising system has been installed at tweendeck level between Nos 2 and 3. Filling and emptying take place through the ballast main in a duct keel which extends forward from the engine room to the after bulkhead of No. 1. The ballast and bilge-suction lines are taken forward through the duct keel and aft through the shaft tunnel. Valves in the ballast, bilge, and fuel systems have pneumatic control. Fresh water is used in the sanitation system.

The combined wheelhouse/chartroom has a good view on either quarter as well as ahead, and the bridge front is recessed so that the automatically operated accommodation ladders can be seen clearly; it is laid out in accordance with the results of work studies. (*Shipp. World* gives a detailed plan.) The auto-pilot can be controlled by the magnetic compass as an alternative to the gyro. The telegraph is of Kwant push-button type.

Pitch controls for the main propeller and the bow-thrust unit (both made by KaMeWa) are provided in the wheelhouse and on the bridge wings.

The bow-thrust unit is driven by a 900-h.p. motor and can produce a lateral thrust of 9½ tons.

Both articles give particulars of the British Paints "Torpedo" coatings (made in Sweden) which were used throughout; various chlorinated-rubber systems were applied to bottom, boot-topping, and topsides. The bottom was finished off with "Torpedo" Epitox anti-fouling, an advantage of which is that the ship can remain dry-docked for several days after it has been applied.

The propulsion machinery consists of two Pielstick PC2V 14-cylinder engines, made by Lindholmens; they drive a 5·7-m (18·7-ft) diameter KaMeWa c.p. propeller through Renk reduction gearing incorporating multiple-disc clutches. The engines are governed to a constant speed of 450 r.p.m. (106·5 propeller r.p.m.) under all conditions. They will normally operate on fuel of up to 1,500 sec Redwood, but provision is made for burning 3,500-sec fuel. Each is rated at 5,300 b.h.p. (normal service) and 5,800 b.h.p. (maximum continuous); the transmission is designed on the basis of a probable increase in m.c.r. to 6,200 b.h.p. (All these power values are metric.) A protective device automatically reduces pitch when the fuel racks or governor settings reach full load. Each engine also drives a 650-kVA ship's service alternator via a flexible coupling and a quill shaft taken through the main-gear pinion. Speed regulation is so effective that the electrical load will not be tripped even during a crash-stop manœuvre. There are also three 340-kW Diesel sets with 1,200-r.p.m. Paxman engines; these are in a special compartment on the after part of the upper flat. In the event of trouble with the main engines or shaft alternators, the non-essential load will be shed and the remaining load transferred either manually or automatically to one of the auxiliary sets. Automatic start-up and shut-down according to load is available for these three sets. Steam-raising plant consists of an automatic Tubox oil-fired boiler and a Sunrod 64S exhaust-gas boiler (this type being considered particularly suitable for a twin-engine installation). Each boiler can produce up to 4,000 lb/hr of dry saturated steam at 75 lb/sq in.

The machinery has been designed for operation without routine watchkeepers; a maintenance staff will work in the engine room during the day. Comprehensive alarm, indication, and recording equipment are provided in the control room, but there is no "data logger". Alarms are also given in the wheelhouse, the engineer officers' accommodation, and the public rooms.

Both articles give general-arrangement drawings and a detailed body plan. *Shipp. World* also gives machinery-layout drawings, a shell expansion, a midship section, and a deadweight scale, together with some information on the *Maihar's* speed and other trials (the actual results are compared with the power speed curves predicted by the Swedish State Tank). *Shipbuild. Shipp. Rec.* compares the power/speed predictions of the Swedish State Tank with those of the N.P.L.

26,890 ***Jag Dev*** **–Prototype "Pioneer" Ship.** *Mar. Engng*, 73 (1968), p. 67 (Aug.) [3½ pp., 2 graphs, 3 phot.]

The *Jag Dev* is the first of their "Pioneer" type Liberty-replacement general-cargo ships to be completed by Blohm & Voss, Hamburg. She is owned and operated by the Great Eastern Shipping Co. Ltd, Bombay.

The " Pioneer " is not so much a class of ship as a representation of a system of shipbuilding which enables the yard to offer a variety of ship sizes and types made up from standard " building blocks " (hull units). Further advantages of the system are that it enables an existing ship, built on these lines, to be " jumboised " or reduced in length if required. Some of the design details and the results of model tests of the " Pioneer " variants were described in Abstracts No. 26,703 (Oct. 1968), 25,483 (July 1967), and 25,223 (Apr. 1967). The principal particulars of the *Jag Dev* are given as:—

Length, o.a.	532·1 ft (162·2 m)
b.p.	496·1 ft (151·4 m)
Beam, moulded	74·8 ft (22·8 m)
Depth, moulded	47·25 ft (14·4 m)
Draught, summer freeboard	34·1 ft (10·4 m)
Deadweight, corresponding	21,600 tons
Grain capacity, maximum	1,049,100 cu ft
Tonnage, gross	13,325
net	9,109
Block coefficient (approx.)	0·74
Speed	16·6 knots
Complement	47

No major problems arose during construction. The absence of plate-forming work, and the fitting of the superstructure as a separate entity during the later stages of fitting out, resulted in considerable savings in costs.

The tweendecks in the cargo spaces are hinged and are operated by the ship's cargo gear; when these decks are closed, flush pontoon covers are fitted in the hatch openings to form a strong tweendeck. The decks have no camber or sheer, which simplifies cargo stowage. All hatch covers on the weather deck, except that to No. 4, which is a deep tank and has a watertight pontoon cover, are of the MacGregor hydraulically-operated single-pull type. There are 16 10-ton derricks mounted on five pairs of unstayed posts, each pair having its own winch platform. All switch-gear for the cargo and topping winches is grouped at the individual winch-control positions.

The ship carries three Spek stockless anchors forward and a smaller one aft. The forward hawse pipes run athwartships to allow the anchors, when being dropped, to clear the bow bulb. The forward anchors are handled by a specially designed Kampnagel windlass.

Propulsion is by an 18-cylinder Pielstick PC2V engine, built by the shipbuilders, and developing 9,000 b.h.p. at 520 r.p.m. In the interests of economy it is intended to burn fuel with a viscosity of 1,500 sec Redwood No. 1; to permit this, special carriers for the top rings have been fitted to the pistons, and the exhaust valves are water-cooled. There is a 5·2 to 1 Renk reduction gear with the thrust bearing incorporated in the gear casing. Between the engine output coupling and the reduction gear is a Geislinger Fawick clutch coupling for use during a crash stop; it is actuated by the main-engine automation system.

The auxiliary equipment is described. There is a 25 ton day capacity Nirex fresh-water evaporator which is heated by the main-engine cooling water. Cleaning and cooling arrangements for the engine lubricating

oil are described, as are the fuel and Diesel-oil systems. The Diesel-oil and lubricating-oil centrifuges can act as standbys for one another. There is a composite boiler having a total evaporation of 1,300 kg (2,866 lb)/hr on the exhaust-gas side, and 1,000 kg (2,205 lb)/hr on the oil-fired side. Some of the other auxiliaries for running the ship's services are described.

A 50-cycle 440-V electrical system is fitted instead of the 60-cycle system originally intended. Power is supplied by three Siemens 330-kW alternators, each driven by a M.W.M. type TbRHS 526A Diesel of 490 b.h.p. at 750 r.p.m.

The results of the sea trials are discussed, and compared graphically with those of the model tests. In the ballast condition and running at 8,640 s.h.p., a mean speed of 16·6 knots and a maximum speed of 16·77 knots were recorded. The contract speed was 16 knots, which could have been obtained with approximately 11 % (1,000 s.h.p.) less power. Crash-stop tests, using the Fawick clutch, resulted in the ship stopping from full ahead in 4 min 45 sec over a distance of about seven ship lengths.

26,891 **" Liberty " Ship Replacements: The " Sante Fe " Class — 19,000-ton d.w. Two-Deck Ship from Euskalduna.** *Motor Ship*, **49** (1968), p. 39 (Apr.) [2 pp., 7 diag., 1 tab.]

General-arrangement drawings, hold cross-sections, and tabulated particulars are given of the first all-Spanish Liberty-ship replacement design, from Cia. Euskalduna de Construccion y Raparacion de Buques S.A., Bilbao.

This is a two-deck all-aft ship, bearing Lloyd's Register classification ✠100 A1, ✠LMC. The principal particulars are: –

Length, o.a.	144 m (472·4 ft)
b.p.	136 m (446·2 ft)
Beam, moulded	22·8 m (74·8 ft)
Depth, to upper deck	13·5 m (44·3 ft)
to second deck	10 m (32·8 ft)
Draught, scantling	9·79 m (32·1 ft)
design	9 m (29·5 ft)
Deadweight, scantling	19,000 tons
design	16,880 tons
Tonnage, gross	10,250
Cargo capacity (grain)	913,360 cu ft
Number of holds	Six
Complement	30

The hull has a straight raked stem and a clearwater stern frame with horn rudder; it is all-welded and longitudinally framed in the double bottom and decks, with transverse framing at the sides. There is no camber or sheer in the decks, but the sheer strake is rounded. The inner-bottom structure is reinforced for the carriage of ore, with small side hoppers to facilitate cleaning the holds. The main transverse bulkheads are vertically corrugated.

No. 3 hold is short, and arranged for both cargo and water ballast; it is suitably located (forward of amidships) to achieve optimum trim and draught with a reasonable bending moment in the ballast condition. No. 5

hold, just aft of amidships, is the largest, allowing stowage of steel bars, etc. A deep tank for fuel has been arranged forward of No. 1.

To facilitate the handling of bulk cargoes, "flap-type" (patent pending) second decks are fitted in Nos 6, 5, 4, and 2. The deck is simply supported on the hatch end beams and girders, and can be raised to form a sloping side. This makes grain boards unnecessary. Nos 1 and 3 have fixed second decks at a lower level than the others; No. 1 second-deck hatch covers can be used as grain bulkheads.

The vessel can carry 267 standard 20-ft containers, 82 of which would be deck cargo. The large second-deck area is suitable for all kinds of vehicles, and is reinforced for the use of fork-lift trucks. Most of the upper-deck hatches, which have single-pull covers, are 12·4 m, i.e. 40 7 ft, wide; the second-deck hatches have steel pontoon covers and are mostly 10·5 m, i.e. 34·4 ft, wide (12·4 m at No. 3). Twelve 5-ton derricks are fitted. The cargo winches and deck machinery are electro-hydraulic.

The standard main engine is a M.A.N. K6Z 70/120D developing 7,200 b.h.p. at 135 r.p.m.; a K5Z 70/120E engine, which develops 8,400 b.h.p. at 140 r.p.m., is offered as an alternative. Two 300-kW high-speed Diesel-alternator sets, and a 1,000 kg/hr composite boiler, are envisaged.

26,892 **The "Freedom" Vessel—How Successful Has It Been?** LASKEY, N. V. *Motor Ship*, 49 (1968), p. 75 (May) [4 pp., 2 tab., 1 graph, 5 diag., 2 phot.]

The Author is a technical director of G. T. R. Campbell (International) Ltd, the designers of the "Freedom" class of Liberty-replacement ships. He refers to certain statements reported in the British press which have criticised, *inter alia*, the design and construction of certain foreign-built Liberty replacements, and refutes them in so far as the "Freedom" class is concerned. About 50 of this class have been ordered, the first completed being the *Chian Captain* (see Abstract No. 26,087, Jan. 1968).

The somewhat sophisticated features of the "Freedom", in comparison with other more conventional replacements being built or on offer, are emphasised. Special reference is made, with the help of diagrams, to the arrangement connecting the side frames to the double bottom, which gives improved stowage for bulk cargoes and vehicles and minimises the danger of grab damage to the structure.

As shown graphically, the results of speed trials in the loaded and ballasted condition for the *Chian Captain* fully confirmed the predictions made from model tests and on which contracts were based; in the loaded condition the full-scale trials showed an improvement in speed over the results of the model tests, and no difficulty has been experienced in obtaining adequate stern trim to give 100% propeller immersion in the ballast condition. Although the model tests and full-scale trial results have shown that the hydrodynamic characteristics of the hull form are good (for a C_b of 0·768 and a L/B ratio of 6·769 to 1), recent model tests with a new hull form have shown even better results. The new-form results are compared with those of the original form in a table; they indicate that about 25% less power is needed with the new form to attain a loaded service speed of 14 knots, and the loss of speed in adverse weather is greatly reduced.

Some trouble has been caused by piston seizures in the Pielstick main engine, which was built under licence by the shipbuilders (I.H.I.). In an

attempt to overcome this, modifications have been made to the oil scraper rings, etc., and the mechanical-ventilation system to the machinery compartment has been improved.

Propeller-excited vibration is not apparent in the " Freedom " ships, as considerable research (which is described) was carried out at Kyushu University to obviate this. The clearwater stern frame accommodating a five-bladed propeller is shown in a diagram; it helps in minimising vibration. A comparison of the fluctuations in vertical and horizontal bearing forces and moments, and in thrust and torque, for a " Freedom " stern, a conventional closed-frame stern, and an AG Weser or Hogner stern is given in a table. The " Freedom " stern is superior on all counts.

Although the first " Freedom " was built in the time specified, a necessary revision of the drive to the propeller resulted in a late delivery. Initial sea trials showed that the Pielstick engine could not be reversed to effect a crash stop from full ahead when in ballast. This failure was due to the fact that the trailing torque of the propeller exceeded the torque which could be produced in the engine by compressed air, and this resulted in the engine stalling. The problem was overcome by integrating a Fawick Airflex clutch drum with the original Geislinger coupling within the space available between the engine and gearbox, the clutch drum being located by axially-oriented coil springs. Details of the assembly are shown in a diagram, which indicates the emergency plate enabling the drive between engine and gearbox to be maintained in the event of failure of the inflatable clutch-actuating tyre. The modified assembly proved successful as it enables the engine to be reversed rapidly and improves efficiency; it is being incorporated in all future " Freedom " ships.

The article includes a photograph of a " Freedom " ship on the building berth at I.H.I.'s Tokyo yard; the groundways incorporate ball-bearings to facilitate movement of the ship during construction.

26,893 *Galila*. **Special Vessel for Rapid Cargo Handling.** *Shipp. World & Shipb.*, **161** (1968), p. 557 (Mar.) [3 pp., 2 tab., 6 diag., 3 phot.]

The ZIM Israel Navigation Company Ltd, of Haifa, have recently taken delivery of a medium-sized general-cargo ship from Orenstein-Koppel & Lübecker Maschinenbau AG, of Lübeck. This ship was originally ordered by the Carib Shipping Company, A S (H. Bang & Co.), Copenhagen. She is an all-aft three-hold two-deck vessel with a raised forecastle. Her principal particulars are:—

Length, o.a.	404 ft (123·21 m)
b.p.	372·4 ft (113·55 m)
Breadth, moulded	54·4 ft (16·6 m)
Depth, to first deck	31·75 ft (9·63 m)
to second deck	22 ft (6·75 m)
Draught, open	21·8 ft (6·63 m)
closed	24·9 ft (7·59 m)
Deadweight, open	5,561 tons
closed	7,027 tons
Cargo capacity, grain	360,937 cu ft (10,222 cu m)
bale	334,258 cu ft (9,465 cu m)
Speed, closed	15·8 knots

The *Galila* has been built to Germanischer Lloyd classification ✠ 100 A4

E1 MC, and to American Bureau of Shipping Class ✠ A1 E Ice Strengthening Class " C " ✠ AMS.

To facilitate rapid cargo handling LMG-Gemini " twin " deck cranes are installed (see also Abstract No. 25,186, Mar. 1967); there is also a single 5-ton crane of similar design forward of the superstructure. The twin crane amidships (between Nos 2 and 3 hatches) is of 2 × 12·5 tons capacity, and there is another, of 2 × 8 tons capacity, between holds Nos. 1 and 2. The jibs of each twin crane can be combined to provide total lifts of 25 and 16 tons respectively. Under certain conditions it will be possible, using the two cranes in combination, to handle loads of up to 40 tons from or into No. 2 hold without having to rig special handling gear.

The large hatchways are equipped with Kvaerner Brug " Trans-Roto " covers on the weather deck and watertight flush-fitting covers suitable for fork-lift truck operation on the tween deck. Two special Olsen-type combination side-door/deck-hatches are fitted on the starboard side, one being between Nos 1 and 2 and the other aft of No. 3. They open upwards (as shown in diagrams), and are actuated and locked hydraulically; each gives a clear opening of 9·8 ft × 7·5 ft (3 m × 2·5 m) in the deck, and 9·8 ft × 7 ft (3 m × 2·2 m) in the shell plating; it is fitted with an adjustable loading platform on which fork-lift trucks can deposit pallets for pick-up by another truck.

The main engine is a M.A.N. type 60/105D six-cylinder two-stroke turbocharged Diesel developing 5,400 b.h.p. at 165 r.p.m. There is no separate control room, but the main engine controls are centralised with 88 alarm points at a soundproof station, enabling watch at sea to be kept by one man only.

Steam is generated in an Aalborg AQ-3 oil-fired auxiliary boiler and an Aalborg AQ-2 exhaust-gas boiler. Electricity is supplied at 200 V, three-phase, 60 cycles, by three 310-kVA Diesel sets running at 600 r.p.m.

Svendborg steering gear is fitted.

A deadweight scale and general-arrangement drawings are given.

26,894 **120 Knots Forecast for Catamaran-Hydrofoil.** *Maritime Reporter*, **30** (1968), p. 6 (15 Sept.) [1 p., 5 phot.]

A twin-hull hydrofoil craft claimed to be the " fastest and most sophisticated vessel in the world today " is being built by Cesco Marine Industries Inc. at their Terminal Island, California, plant. This prototype, designated *Cescat I*, was designed using a combination of marine and aerodynamic engineering concepts. The twin hulls are of aluminium alloys Nos 5456, 5086, and 5454, and are characterised by straight lines except for the sweep of the stem and chine lines up to the stem head. The overall length is 80 ft, and the overall beam 24·2 ft. The light-ship weight is 49 tons.

The fully-submerged hydrofoils are of patented design and are supercavitating over the whole foilborne speed range. They will be operated hydraulically and controlled by wave sensors. The stilt height is such that the vessel will be able to " fly-over " state 3 to 4 seas at top speed.

Design speed when foilborne is 120 knots, and the hullborne (planing) speed is estimated to be 75 knots. In both cases the aerodynamic-wing design of the centre body tends to stabilise the vessel on an air cushion, and to eliminate wave slapping.

Propulsion can be by waterjet or by turbofan engines. The *Cescat 1* is powered by one modified aft-mounted J-97 turbofan engine on each hull; the modification has reduced the specific fuel consumption to 0·30–0·35 lb/h.p.-hr. Replacement of an engine can be done in four hours.

The builders feel that there are many applications for craft of this basic design in both military and commercial activities. The article includes photographs of *Cescat 1* under construction.

INDUSTRIAL AND ECONOMIC INFORMATION

26,895 **British Shipbuilding Costs: September 1968.** *Motor Ship Special Survey on " British Shipbuilding Today "*. Sept. 1968: p. 55 [4 pp., 6 tab.]

The article gives the estimated costs of building the following ships in the U.K.: (*a*) a 13,150/15,300-ton d.w. shelter-deck cargo ship, (*b*) a 72,500-ton d.w. bulk carrier, and (*c*) a 102,000-ton d.w. tanker. The particulars of these ships are listed and the detailed cost analyses given; those for November 1967 and for September 1968 are compared in adjacent columns.

All the vessels are of the designs used in the last analysis (see Abstract No. 25,992, Dec. 1967). Specifications of systems and equipment are included.

The totals for September 1968 are:—

	(*a*) £	(*b*) £	(*c*) £
Hull and outfit .	711,930	2,012,200	2,918,490
Machinery . .	351,890	699,090	1,027,160
Other charges .	164,000	394,000	510,000
Total cost of ship	1,227,820	3,105,290	4,455,650
	(1,180,275)	(2,982,410)	(4,261,700)

The totals in parentheses are for November 1967.

Although labour costs have not changed since last year, because there have been no national labour increases, it is pointed out that the national wage agreement being negotiated may have a considerable effect on future labour costs. Also, the full effect of devaluation may not yet have been realised, and it is possible that subcontractors' prices may now increase due to the demand following the recent increase in orders. The price of steel rose about 2% owing to devaluation, and foreign sub-contracted items, wood, and paint have been particularly affected. The present state of the industry is nevertheless encouraging; some data are quoted to illustrate this.

26,896 **[Examples of] Capital Investment Analysis—A [Computerised] Simulation Approach.** MAKADOK, S. *A.S.M.E., Paper No. 67-PEM-7, presented* 10–12 Apr. 1967 [7 pp., 1 ref., 3 tab., 4 diag.]

By modelling present and proposed systems on a computer and varying the controlling parameters, a cost-effectiveness relationship may be established before committing funds. Three actual applications of the technique are described, with flow diagrams and typical print-outs.

These are (*a*) choice between coal, gas, and oil fuel for an electric power plant capable of burning any of them; (*b*) choice of printing method for a department producing technical and trade literature; (*c*) design of a water-supply system with rapid fluctuations of supply and demand.

AIR-CUSHION VEHICLES

(*See also Abstract No.* 26,865)

26,897 **Research on Hovercraft over Calm Water.** EVEREST, J. T., and HOGBEN, N. *Trans. R.I.N.A.*, **109** (1967), p. 311 (July) [12 pp., 12 ref., 1 tab., 24 graphs, 4 diag.; and Discussion: 4 pp., 3 ref., 1 tab., 1 graph, 2 diag.]

Research on the hydrodynamics of hovercraft has been in progress in the Ship Division of the National Physical Laboratory for some years. Up to the middle of 1965 it was entirely concerned with calm-water performance, and may broadly be divided into two phases. The first phase covered experiments with a circular model mounted at fixed attitudes, and also some work on a two-dimensional fixed-attitude jet. The results of this research have already been reported (see, *inter alia*, Abstracts No. 22,604, Feb. 1965, and 22,977, May 1965), but a resumé is included in the present paper. The paper is, however, mainly devoted to the second phase, which has covered experiments with a freely towed rectangular model. Some of the results obtained over deep water and over shallow water are reproduced (see also Abstracts No. 24,981 and 24,982, Jan. 1967).

For deep water, there are graphs showing variations of drag, trim angle, and c.g. rise with forward speed and Froude number; also the effects of model weight and speed on drag, trim, and rise height. Measurements of the craft-generated wave pattern were used in drag estimation by Egger's transverse-cut method (see Abstract No. 25,195, Apr. 1967); the results, which are shown graphically, agree well with the theory published in 1965 by M. J. Barratt. For shallow water, wave-drag theory is compared with results derived from the trim records, for three water-depth/cushion-length ratios. The theory used appears to be unsatisfactory in shallow-water conditions. However, an attempt is made to define an empirical relation between wavemaking-induced trim and Froude number.

A breakdown of measured hovercraft resistance into the following components is suggested: trim drag; inlet momentum drag; outlet momentum drag; aerodynamic profile drag; wetting, spray, and water-contact drag. Finally, non-steady conditions (acceleration, transition from deep to shallow water) are discussed.

The paper is concerned entirely with unskirted craft. Many of the basic principles are, however, also valid for skirted craft, and it is mentioned that, from 1965 onwards, N.P.L. have been model-testing such craft with the aid of specially-developed equipment.

SHIPYARDS, DOCKS, AND PRODUCTION METHODS

26,898 **PERT as an Aid to Building Ships.** PLEDGER, M. N. *Marine Systems*, **2** (1967), p. 71 (July/Aug.) [2 pp., 1 diag.]

This article is based on a paper read by the Author at the Network Planning Users Conference, London, June 1967.

The Author describes PERT planning at Alexander Stephen & Sons, Linthouse (*Editor's note:* now part of Upper Clyde Shipbuilders Ltd) whose constructions ranged from dredgers to warships. He outlines the facilities for PERT available at this yard at the time of writing, and the training involved. With the introduction of PERT in 1967 the firm's production organisation was reorganised; this is illustrated with a diagram.

On receipt of contract n overall network was prepared covering all stages from design to acceptance trials. It could have between 500 and 1,000 activities, the number being kept as low as was consistent with clarity. Emphasis was placed on design activities and purchasing for principal items of bought-out material, because 80% of a ship's cost and also the possible delivery date depend on sub-contractors. The overall network was used to monitor progress, to give early warning of potential delays, to schedule drawing-issue and materials supply dates, to verify the feasibility of " contract milestones ", and to gauge the effects of proposed design changes. Critical paths were examined and precautions taken to safeguard ship delivery dates.

In addition to the overall network, a cargo ship was usually divided into four groups of sub-networks, concerned respectively with machinery installation, accommodation outfitting, deck and hold equipment, and electrical systems. These could be subdivided to keep the number of activities per network in the range 500–1,000. All the sub-networks were related by interface networks lifted from the overall network, so that interdependencies were not lost. Details of the procedures adopted within each group are given. It is in accommodation fitting-out that network planning has offered the greatest advantages.

Network analysis is mandatory in warship building, the main sub-networks being (i) Machinery installation, (ii) Systems. In this case the overall network is much larger than in that of merchant ships; it consists of between 1,500 and 2,000 activities.

26,899 **Reducing Construction Time by Mass-Production Techniques.** *Maritime Reporter*, 30 (1968), p. 40 (1 Sept.) [1 p., 1 diag., 2 phot.]

The Author outlines the introduction of mass-production techniques (as used in the automotive and aerospace industries) by General Dynamics at its Quincy, Massachusetts, shipyard. Although the new system is not yet in full operation, it is already saving tens of thousands of man-hours. It is flexible enough to result in high production efficiencies for non-series as well as series ships.

Aspects of the system include: a production-line approach to simultaneous construction of different classes of ships; the fitting-out of large sub-assemblies with machinery, piping, etc., before incorporating them in the ship (this is made possible by precise pre-production design and planning and has already cut fitting-out man-hours for some sub-assemblies by more than 40%); improved welding processes that increase production flow; automated equipment to cut basic steel shapes; and a special computer program to reduce the time spent in converting design into construction.

An important feature of the new system is an automatic shot-blasting facility (shown in a diagram) which can clean sub-assemblies weighing up to 75 tons in a tenth of the time formerly required. The sub-assemblies

are moved into and out of the blasting chamber on trailers. During blasting, they are suspended from, and rotated by, a special 75-ton crane. Cleaning is performed by 32 centrifugal blasting wheels, each of which throws 500 lb of pellets per minute. Average cleaning time is 25 minutes. After cleaning and inspection, any areas which were shielded from the automatic process are cleaned manually by air-hose shot-blasting. The facility has been in operation since April 1968 and is believed to be the world's largest airless blast-cleaning plant. It was designed in co-operation with Wheelabrator Corporation, Mishawaka, Indiana, who also provided the systems for handling and recirculating the abrasive, and for dust removal and ventilation. The use of this facility means that steel can be received in mill condition, thus cutting costs (the yard's steel was formerly pre-painted by the vendor). On being drawn from the stockyard steel is locally cleaned in the vicinity of subsequent markings, cuts, welds, etc.

The shot-blasting facility will soon be complemented by a controlled-climate paint shop constructed alongside it; the two together will form the world's first all-weather preparation and painting shop for large prefabricated units.

26,900 **Avondale Shipyards "Rolls 'em Over".** *Mar. Engng*, 73 (1968), p. 64 (Aug.) [2 pp., 4 phot.]

The initial phase of an extensive mechanisation programme to increase production of multiple units has been completed at Avondale Shipyards, New Orleans, Louisiana. Concurrently with this programme the yard has developed its design side by recruiting additional technical staff and introducing computers and automated yard machinery.

Steel plate and sections arrive by ship, rail, or barge, and are handled in a 240-ft wide storage area by a massive stock crane equipped with a magnetic beam. Selected plates are lifted by a travelling overhead hoist and deposited on to a tracked "captivator" by sets of magnetised rollers. The plates are then passed through a rotary shot-blasting machine, a washing device, a drying oven, a priming-paint spraying machine and, finally, a second drying oven; they are then lifted by an electro-magnetic crane on to cutting platens in the plate shop ready for numerical marking, cutting, and forming. The cutting is performed by portable three-axis machines each fitted with three torches for edge bevelling, and capable of preparing simultaneously up to four 8-ft wide plates.

After preparation the plates are moved to the prefabrication area, where they are incorporated into modules (a method of construction first used by Avondale in 1942). The completed modules are moved on wheels to the assembly lines, of which there are two, one for large merchant ships and the other for naval craft; both assembly lines are arranged for broadside launching. Building progress is synchronised by computer, as is the final fitting out. At present one destroyer escort can be launched every seven weeks.

To permit a maximum amount of downhand welding, the completed modules for the destroyer escorts (except for the bow and stern units) are positioned on the assembly-line platen in an inverted position prior to being joined. After joining, the hull unit is lifted hydraulically from the platen and moved laterally into a set of powered turning rings, locked in position, and turned upright; this whole operation takes less than six

hours. When upright, parts of the turning rings are removed, allowing the hull to be moved sideways ready for the fitting of the bow and stern units. After further lateral movement, the prefabricated superstructure is fitted and the ship is ready for launching. Some of the stages in the building procedure are shown in photographs.

26,901 **Handling Long Loads in the Shipyard by Means of a Large Self-Propelled Carrier** (in French). MARTIN, J. *Tech. Mod.*, **59** (1967), p. 446 (Nov.) [2 pp., 1 tab., 1 phot.]

The straddle carrier described, the " Mammouth ", is intended for a shipyard in northern France, where it will be used for moving plate and section material from the point of arrival to the stockyard and from the stockyard to the appropriate workplace. It can handle plate up to 15 m by 2·5 m (49·2 ft by 8·2 ft), and weighing up to 15 tonnes. The plates are carried, in the horizontal position, by means of a number of electromagnets suspended from a frame which is itself suspended (by the lifting cables) from the top of the straddle carrier; the frame is provided with vertical guides to prevent it swinging, and the magnets have shock-absorbing mountings. There are 40 magnets, but it is not necessary to energise all of them when the smaller plates are carried. Sections are carried, in batches, in special containers suspended from the top of the straddle carrier in place of the frame. Lifting and lowering the frame or container are done conventionally, a hydraulic winch for this purpose being mounted in the top of the vehicle.

Each of the carrier's four wheels is steerable, and two of them are driving wheels. Hydraulic power for lifting, steering and driving, and electric power for control, are supplied from an 80-h.p. (metric) Diesel engine driving a variable-delivery pump and a 6-kW generator; this equipment is mounted in the top of the vehicle. The hydraulic-system pressure is 200 bars (2,900 lb/sq in). The control-cabin mounting allows it to be moved as required, either outboard to provide clearance for loading and unloading or inboard to reduce overall width for passing through doorways, etc. The four steerable wheels permit crabwise movement (up to 90°), and would also allow " on the spot " turning; the Author (of the firm of Ferrand & Frantz) describes the patented steering system in some detail.

The main framework is of bolted box-construction; the chassis " suspension " consists simply of side beams, free to pivot at their mid-points, so that the wheels at each end of the beams are all kept in contact with the ground; each wheel has a Boeing 707 tyre.

The article gives further information on the construction and control of the Mammouth. Principal particulars include:-

Length, o.a.	7·5 m (24·6 ft)
Breadth, o.a.	6 m (19·7 ft)
Height, o.a.	6 m (19·7 ft)
Wheel track	4·85 m (15·9 ft)
Wheel base	5 m (16·4 ft)
Under-hook height	2·9 m (9·5 ft)
Turning radius	4 m (13·1 ft)
Hoisting speed	0 to 5 m/min (0 to 16·4 ft/min)
Translation speed	0 to 5 km/hr (0 to 3·1 m.p.h.)

26,902 Portable Machine for Cleaning Shot-Blasting Abrasives. *Engrs' Digest*, **28** (1967), p. 111 (Nov.) [½ p., 1 phot.]

The Penfold Metallising Co. Ltd have developed a portable machine, the " Reclaim " separator, for cleaning both metallic and non-metallic shot-blasting abrasives of any type whose basic particle size is not less than 100-mesh. The machine, which enables the abrasive to be recycled repeatedly, has the important secondary function of raising the abrasive from near ground level to reload the blasting machine with dust-free material, without manual effort. It is operated by air at a pressure of 25 lb/sq in at the rate of 80 cu ft/min; non-metallic grits can be cleaned and cycled at 1 ton/hr and chill-cast iron or steel shot at 2 tons/hr.

MATERIALS: STRENGTH, TESTING, AND USE

26,903 Factors in the Choice of Materials for Marine Engineering. TODD, B. *Inst. Mar. E., paper read* 23 Jan. 1968 [12 pp., 33 ref., 11 tab., 1 diag., 4 graphs, 2 phot.]

The metallurgical problems encountered in marine engineering arise not from the availability of suitable materials but largely from the industry's requirement of long life—normally about 20 years—and because of the large size of some of the components that the metal-workers have to provide.

The Author has divided his paper into sections, each of which deals with a component or group of components, and the material requirements for each. He starts by analysing the causes of engine-room and propeller failures he investigated over a five-year period, all of which were associated with mechanical stress. This analysis shows that, of 171 failures investigated, brittle fracture caused 2·5%, fatigue and corrosion fatigue 68·5%, stress-corrosion cracking 14·5%, and creep and other high-temperature effects 14·5%. The outstanding importance of fatigue and corrosion fatigue as the main modes of failure is, in the Author's opinion, even greater than these figures indicate because of certain restrictions imposed by the conditions under which the failures were investigated.

Structural Components. These comprise parts such as bedplates, entablatures and gearcases, for which steel fabrications are now widely used, especially in the larger sizes. Difficulties encountered in early fabrications due to the low fatigue strength of weldments can be avoided by machining off the weld reinforcement and polishing the weld, provided that there are no internal defects in the weld. The fatigue strength can then be as high as that of the rolled plate with mill scale left on. Stress relieving may sometimes be beneficial—it is essential in gearcases, for example—but the avoidance of stress raisers such as poorly-profiled weld reinforcements is of equal importance.

Diesel Engine Crankshafts are now usually built up of a number of forgings or of forgings and castings. The shaft forgings used to join crank webs or throws together are commonly made of vacuum-degassed steel, which ensures good mechanical properties and smaller risk of defects, in particular hydrogen cracking, in the forgings. A point to be noted is that the fatigue limit in air, which for plain unnotched bars is about 50% of the ultimate strength, may be appreciably reduced in the larger sizes. In one

particular steel the fatigue limit was reduced by 27 to 29% when the bar diameter was increased from ⅜ in to 2 in.

Many crank throws and crank webs are still made by casting, and defects may occur when, for example, the casting requires the pouring of over 50 tons of steel. Magnetic and ultrasonic methods of non-destructive testing can readily detect large defects. Fine porosity, which often appears during final polishing of the crankpin, is not a cause for rejection of the casting provided that the porosity is well dispersed and that ultrasonic testing shows the underlying metal to be sound. Weld repairs of castings may lead to fatigue failures if not properly executed; and if the original defect on a large casting is relatively small it may be safer to grind it out, leaving a shallow depression, than to incur the risk associated with an unsatisfactory repair by welding.

Cylinder Components. Cast-iron pistons are liable to failure due to creep on the hot face under the compressive thermal stresses associated with normal running, and the reversal to tensile stress when the engine stops. Continued cycling on starting and stopping leads to fatigue failure. Steel is more resistant to this type of failure, and various steels are used. Molybdenum and chromium-molybdenum steels have high creep strength, and a 13% chromium stainless steel, which also has high scaling resistance, is used when burning of piston crowns is a problem.

For cylinder covers, carbon steel or chromium-molybdenum steel is normally used in the larger engines as thermal loading increases, although cast-iron covers are satisfactory for small engines.

Cylinder liners may crack under the severe thermal stresses caused by high wall temperatures; but these failures are not due to fatigue, and cast iron is the material most commonly used because of its high resistance to wear. For large marine engines a vanadium iron is normally used; a table shows that the addition of 0·08 to 0·14% of vanadium reduces the wear rate, as compared with vanadium-free iron, by upwards of 50%.

Turbine Components. Creep resistance and ductility are important at temperatures above about 340° C (650° F). Molybdenum, chromium, and vanadium improve the creep resistance of steel. Weldability is also important, and the carbon content is usually kept low. Turbine casings, valve chests, and other cast components are usually homogenised at 850–950° C, normalised, and tempered at 600–650° C. For turbine rotors, steels low in chromium and manganese are recommended, while for turbine blading stainless iron (0·15% carbon and 13·0% chromium) is normally used. It has good corrosion resistance in wet steam, and creep resistance that enables it to be used up to 480° C. For bolted joints in high-temperature steam piping and valves, Nimonic 80A has better properties than the chromium/molybdenum and chromium/molybdenum/vanadium steels commonly used which may fail above 500° C by creep or by embrittlement.

Marine boilers are liable to suffer from high-temperature scaling due to the vanadium and sulphur contents of heavy fuel oil, and the sodium present in the marine atmosphere. The excellent resistance shown by 50/50 and 60/40 chromium/nickel alloys has led to their widespread use in marine boilers.

Gear materials should have high fatigue strength to resist tooth breakage, and wear-resistant surfaces to resist wear, scuffing, and pitting. Steel with an ultimate tensile strength of about 55 ton/sq in is suitable if surface

hardened either by case-hardening or induction hardening. Tests of gears carburised to 1·1–1·3% carbon showed greater pitting resistance than those of gears of similar hardness with 0·7–0·8% carbon. For pinions and wheel rims En 25 (2½% nickel/chromium/molybdenum steel) has been used with great success; it has high resistance to pitting and scuffing.

Corrosion. The Author discusses the various types of corrosion that can be caused by sea water, with particular reference to velocity and galvanic effects, and also to stress-corrosion cracking, dezincification, and graphitisation. He also summarises the corrosion behaviour of a number of materials used in marine engineering. This leads to consideration of the choice of materials for the components of cooling systems, among which he includes: heat-exchanger tubing, tube plates, water boxes and end covers, pumps, and valves and filters. Suitable materials for all these are indicated.

Propellers need materials of high tensile and proof strength, and good stress-corrosion and corrosion-fatigue resistance; they should also be easy to cut, weld, and machine. For the largest propellers aluminium-bronze alloys are satisfactory but expensive; for smaller propellers cheaper materials such as manganese-bronze, stainless steels, and sometimes cast iron can be used. The properties of a number of propeller alloys are tabulated.

26,904 **Low-Cost Portable Crack Detector.** *Works Engineering & Factory Services*, **63** (1968), p. 12 (Apr.) [1 p., 2 phot.]

The Amlec crack detector, developed at the Admiralty Materials Laboratory and exploited by the N.R.D.C., is manufactured under licence by Inertia Switch Ltd, and D.E.B. Electronics Ltd. It is a light-weight instrument powered by a battery and fitted with both visual and audible indicators. Using eddy-current technique, it enables surface cracks in ferrous-metal components to be easily and rapidly detected.

Two types of probe are available, one for flat plates and the other for internal angles less than 90°. A unique feature of the system is its independence, once the circuit has been balanced for a particular specimen, of the gap between probe and material. As a result, it can be used on rough surfaces, which may be covered with rust, paint, scale, or dirt. Experienced operators can also, after calibration of the instrument, estimate crack depth and structure.

BOILERS AND STEAM DISTRIBUTION

26,905 **The Babcock Bi-Drum Boiler.** *Shipp. World & Shipb.*, **161** (1968), p. 565 (Mar.) [2 pp., 2 diag.]

The main features of this new boiler, designed and developed by Babcock & Wilcox (Operations) Ltd, are: low production and reduced erection costs, the elimination of brickwork, long flame travel, and simplified cleaning.

Basically the boiler consists of two connected parts separated by a screen wall. The first part is a fully water-cooled furnace of optimum proportions, with the oil burners mounted in the roof. The combustion

products pass down the whole height of the furnace; this results in long flame travel and extended furnace residence, which ensure complete fuel combustion with a minimum air/fuel ratio. The second part consists of a chamber containing the superheaters and main bi-drum bank. The superheating surfaces consist of primary and secondary sections, each being formed of horizontal multi-loop elements. Provision is made for parallel steam and gas flow, giving a minimum tube characteristic consistent with an acceptable steam-temperature characteristic at the desired level. The main bi-drum bank consists of two groups of tubes which are connected to the steam drum at the top and the water tubes at the bottom.

After combustion, gases flow from the furnace via the open lower portion of the screen and turn in a cavity before flowing upwards over superheater and economiser surfaces. On leaving the boiler the combustion gases may pass to an economiser and/or a gas/air heater. A steam/air heater may also be incorporated.

Final steam temperature is controlled by interposing an attemperator (with a bypass) between the primary and secondary stages of the superheater. Steam flow through the attemperator is controlled by a two-way valve or two separate valves, the regulation of which is influenced by the steam-temperature control equipment.

In the Bi-Drum boiler Babcock & Wilcox have made full use of the membrane-wall construction which they introduced into the U.K. In this, longitudinally-finned tubes are welded together forming a fully water-cooled gas-tight panel; such panels form the enclosure and screen-walls of the furnace and convection-surface chamber. This simplifies erection and eliminates the necessity for a separate gas-tight steel casing. Air-pressurised seals are provided where penetrations are required for soot-blowers, etc. However, the whole unit can be double-cased and the space between casings pressurised by combustion air. (In this scheme the outside of the membrane walls is not insulated.) Instead of troublesome firebrick quarls, water tubes are bent closely around the burner openings.

Operating methods are as for any other marine boiler. All control equipment is conventional and meets the recommendations of Lloyd's Register on Automatic Controls in Ships (see Abstract No. 25,363, May 1967). Normal feed-water treatment is satisfactory provided that the specified limits of purity are strictly maintained.

The first Bi-Drum boiler has been installed in the tanker *Esso Mercia* (see Abstract No. 26,886, this issue); its operating conditions are stated in a table. It is designed to evaporate 210,000 lb/hr at 900 lb/sq in and 955°F (superheater outlet), together with 35,000 lb/hr at 500 lb/sq in and 625°F (desuperheater outlet). Boiler efficiency is given as 90.7%. The four burners are of Racer Y-jet type (see Abstract No. 25,913, Nov. 1967), with a turn-down ratio of 18 : 1.

DIESEL AND OTHER I.C. ENGINES

26,906 **Prediction of Wall Temperatures inside Diesel Engines by Digital Computer.** Wu, H. L. *Engr*, **222** (1966), p. 789 (25 Nov.) [5 pp., 8 ref., 1 tab., 12 graphs, 5 diag.]

In order to make accurate design-stage predictions of temperatures,

heat flows, and thermal stresses in cylinder walls and heads, a step-by-step integration for the complete engine cycle is necessary, taking into account the continuously varying nature of the heat source. In the work described, a computer was used for this purpose. A simplified block diagram of the program is given, together with an explanation of the steps involved. The program incorporates equations relating each of the following stages of the heat-transfer process, which are discussed in turn:

(1) Rate of heat release from the combustion gases inside the cylinder; this depends on engine load, speed, and type of fuel injection. Its cyclic variation curve has a sharp narrow peak in high-speed engines, which becomes progressively flatter and broader in medium- and low-speed engines; this is illustrated by a series of graphs calculated by computer from cylinder-pressure diagrams. On this basis, algebraic expressions adequate for computer performance predictions are given for rate of heat release, and fraction of total heat released, at each instant of combustion in lightly-loaded and heavily-loaded medium- and low-speed engines.

(2) Heat transfer between combustion gas and cylinder walls. This is directly dependent on the temperature, pressure, and movement of the gas inside the cylinder and the degree of radiation from the combustion flames. Although these quantities fluctuate during the cycle, the wall surface temperature at a fixed point on the cylinder wall can be considered constant because the cyclic fluctuation of the metal surface temperature is known to be small.

(3) Heat conduction through the walls. It is necessary to take account of the convergence and divergence of the heat flow paths through the walls as these have a direct influence on the temperature gradients. This is especially so for intricately-formed cylinder heads.

(4) Heat transfer from the walls to the cooling water. The rate of heat flow from the walls to the water is dependent on water flow velocity and the extent of boiling of the water at the contact surface. (The occurrence of nucleate boiling results in better heat transfer and causes a marked discontinuity in the curves of wall temperature against engine load.) Also, the formation of scale deposit on the walls obstructs the heat flow.

Computer time is about three minutes for determining the temperature pressure gas cycle for given inlet conditions in a given engine (mean and maximum pressures being specified), and less than one minute for the heat transfer through any particular point of the wall. Some results of such calculations for a supercharged two-stroke uniflow-scavenged engine (bore 290 mm, stroke 400 mm, 500 r.p.m.) are presented graphically and discussed; they illustrate the influences of water flow velocity, thickness of scale deposit, cylinder-head thickness, and convergence of heat-flow path through the wall material. The temperature calculations showed marked peaks on the gas-side surface between adjacent exhaust-valve seatings; this, and the occurrence of boiling at certain points of the cylinder head, were confirmed by thermocouple measurements on an actual engine. Another comparison between prediction and measurement, for a highly turbocharged four-stroke engine (300 mm bore, 400 r.p.m.) showed that the calculation method is applicable to this type also.

26,907 **Photoelastic Investigation on a Turbocharger Rotor** (in German). Nitt, B. *M.T.Z.*, **29** (1968), p. 414 (Oct.) [8 pp., 12 ref., 9 graphs, 10 diag., 5 phot.]

POWER TRANSMISSION

(*See also Abstract No.* 26,913)

26,908 **Influence of Hull-Structure Stiffness on the Magnitudes of Shaft-Bearing Reactions** (in Russian). KVASHUK, N. F., KONTOROVICH, B. M., and ZENOVA, I. A. *Sudostroenie*, No. 9 (1968), p. 15 (Sept.) [4 pp., 1 ref., 3 tab., 1 diag.]

In shafting design calculations, hull structure has hitherto been considered to be rigid. However, the construction of large tankers and multicargo vessels, with powerful propulsion machinery and high propeller thrust, has led to a sharp increase in shafting diameters, and shaft stiffness is still further increased by the aft location of the main machinery.

Reliable operation of the propeller/shafting/thrust-bearing system can only be secured by ensuring that the bearings, particularly the stern-tube bearings, are always loaded, since the natural frequency of the system's lateral vibrations is then sufficiently remote from the exciting frequencies. If any of the bearings, particularly the stern-tube bearings, lose contact, the frequency of the system falls and the danger of blade-frequency resonance arises.

A bearing may lose contact through various causes; e.g. wear-down of stern-tube bushing, or shifting forward of the effective support point in the after stern-tube bearing.

In this paper an analysis is given of the possibility of loss of contact upon a change in bearing reactions, due to deflections of the hull structure in a large tanker now being designed. The stern and shafting of this vessel are shown in a diagram.

There are five points of support for the shaft: (1) the stern-tube bearing, (2), (3), and (4) intermediate bearings, and (5) the thrust bearing. The vessel was considered in the following conditions: in ballast in still water with full afterpeak and with empty afterpeak, in ballast in the trough of a wave with full afterpeak and with empty afterpeak, fully loaded in still water, and fully loaded on the crest of a wave. The wave height considered is 6 m (19·6 ft).

The propeller/shafting system is subject to static forces (weight of propeller and shaft) and dynamic forces caused by the action of the screw. Reactions of the shaft bearings were determined by static continuous-beam calculations, while the dynamic forces on the propeller were calculated as their amplitudes of variation.

Deflection of the hull structure was calculated at each shaft bearing. The types of deflection examined are: overall bending of the stern; bending of the bottom structure; longitudinal shifting and tilting of the thrust block due to propeller thrust.

A comparison of the values of bearing reaction, derived by static calculation, with the total change of reaction, due to deflection of the hull structure, shows that the influence of this reaction change is insignificant and is noticeable only at support points (4) and (5). This comparison is tabulated, as is the total deflection of the hull structure under the shaft bearings, and the relative amount of each type of deformation involved. For support points (4) and (5) these relative amounts are: 70% by the tilting of the thrust block, 25% by bending of the bottom framing, and 5% by hull bending.

As the tilting of the thrust block appeared to have the greatest effect on

altering the bearing reactions, experiments were carried out on a tanker at present in service. The results of the experiments confirmed, in the main, the theoretical results.

With the present method of hull design the overall stiffness of the after part and of the bottom structure has no substantial effect on bearing reactions. However, the longitudinal shifting of the thrust block due to propeller thrust does have a substantial effect on the frequency of the longitudinal vibration of the propeller/shafting/thrust-bearing system. In designing the seating of the thrust block and the construction of the tank top in this region, it may be necessary to increase the stiffness, by increasing the depth of the double bottom and by increasing the extent of the seating and the intercostal girders which support it.

MACHINE PARTS

(*See also Abstract No.* 26,903)

26,909 **The Axial Stiffness of Marine Diesel-Engine Crankshafts. Part 1—Comparison between the Results of Full-Scale Measurements, and those of Calculations According to Published Formulae.** VISSER, N. J. *Int. Shipbuild. Progress*, **14** (1967), p. 452 (Dec.) [14 pp., 7 ref., 14 tab., 5 graphs, 4 diag.]

This is Report No. 102M of the Netherlands Ship Research Centre TNO.

The Author first describes how complicated vibration phenomena may occur in the shafting systems of ships having large-bore low-speed direct-coupled main engines, especially when the shafting is short. These phenomena may produce inadmissible torsional and axial vibrations at speeds within the manœuvring range of the engine. The results of earlier work on predicting the natural frequencies and modes of coupled torsional/axial vibrations at the design stage of an engine are mentioned (see also Abstract No. 21,119, Feb. 1964).

In this Report the various methods used in measuring the axial stiffness of the cranks of four full-scale marine crankshafts, stationary and supported by the engine main bearings, are described. The characteristics of each crankshaft, and the methods of applying the loads, are described with the help of diagrams and a data table. The measurements obtained are presented graphically and summarised in tables. The results are discussed and compared with those of calculations when using six published formulae.

The relationship between the coefficients and the angles of adjacent cranks, or the " mean " crank angle, is investigated. The experimental results are verified by the theoretically-determined shear-force moments for axially-loaded adjacent cranks, and the experimentally and theoretically-determined stiffnesses are compared. None of the published formulae can be considered reliable. The calculated results for the one fully-built crankshaft tested are considerably worse than those for the other three (semi-built) crankshafts.

Conclusions are given and proposals made for future work (see following Abstract).

26,910 **The Axial Stiffness of Marine Diesel-Engine Crankshafts. Part 2—Theory and Results of Scale Model Measurements and Comparison with Published Formulae.** LINDEN, C. A. M. VAN DER. *Int. Shipbuild. Progress*, **15** (1968), p. 96 (Mar.) [9 pp., 8 ref., 7 tab., 3 graphs, 3 diag.]

This is Report No. 103M of the Netherlands Ship Research Centre TNO.

The Author refers to other work on this subject, covered in the preceding Abstract.

The present Report describes a method of determining axial elasticity by means of measurements on 1 : 15 scale models of individual cranks of a six-, nine-, or ten-throw shaft. The cranks, which are shown diagrammatically, are clamped at one end and loaded by moments and forces at the free end. By measuring the displacements and rotations, a set of influence numbers can be obtained which are used for calculating the axial elasticity for the whole crankshaft, and also for its individual cranks. The position of the crank in the shaft considerably affects its elasticity, but by using the influence numbers this can be taken into account, both for rigid and elastic bearings. A simple formula for the elasticity of a crank is presented; a more accurate computer method can also be used. The results of calculations, which are shown in tables and graphs, are compared with those according to published formulae and are discussed.

It is concluded, *inter alia*, that from a loaded scale model it is possible to obtain all the data necessary for calculating the performance of the crankshaft under all conditions. However, further research into the elastic properties of bearings is necessary.

26,911 **Low-Cycle Fatigue of Large-Diameter Bolts.** SNOW, A. L., and LANGER, B. F. *A.S.M.E., Paper No. 66-Pet-8, presented* 18–21 Sept. 1966 [8 pp., 12 ref., 4 tab., 13 graphs, 2 diag.]

New fatigue data on high-strength bolting materials (in particular AISI 4340 steel), under axial loading at constant strain or deflection amplitude, are presented and analysed together with previously published data. Two test temperatures are involved, viz., 70° and 500° F. The relationship of cyclic to static yield strength is examined. The results of several series of fatigue tests, at constant load amplitude and the same two temperatures, on full-size (1 to 3·25 in shank diameter) studs are then analysed; a comparison with the basic fatigue-strength curve for the unnotched material gives experimental fatigue-strength reduction factors for bolts and studs. The variations of these reduction factors with thread root radius, stud size, thread taper, and other variables are studied. A new fatigue-design procedure, involving a proposed design curve which takes account of the additional information now available, is explained.

It is found, *inter alia*, that the bolting-fatigue design rules of the ASME Boiler and Pressure Vessel Code, Section III (Rules for Construction of Nuclear Vessels) are too conservative.

26,912 **Cyclic Stress for [High-Strength] Bolts and Studs.** FRITZ, R. J. *A.S.M.E., Paper No. 67-MET-23, presented* 3–5 Apr. 1967 [21 pp., 23 ref., 3 tab., 8 graphs]

Fatigue data for high-strength fasteners under axial loading are examined and compared with ASME Boiler and Pressure Vessel Code (Section III) rules. These data include 198 martensitic bolts and studs with the following properties: size 1 in to $5\frac{1}{16}$ in; AISI steels 4140 and 4340; ultimate strength 130×10^3 to 170×10^3 lb/sq in; unplated and plated with copper, silver-on-nickel, silver, chromium, or cadmium; rolled, ground, or machined-and-ground threads; machined and machined-and-rolled bolt-head fillets. The approximate cyclic range of failure was 3×10^3 to 5×10^5. The tests were at room temperature and at 500° F.

Other available notched fatigue data are also examined, with special reference to the effect of mean stress. New cyclic-stress limits are proposed on the basis of the data presented, and various other design recommendations are made.

MARINE POWER INSTALLATIONS (GENERAL)

26,913 **The Selection and Application of Medium-Speed Machinery Systems.** ADLEY, A. A., and LEA, K. E. *Motor Ship*, **49** (1968), p. 27 (Apr.) [6 pp., 3 tab., 12 graphs, 2 diag.]

This is a condensed version of a paper read to the Inst. Mar. E. (South East England Branch) on 15 Feb. 1968. It deals with three main topics, viz: the adaptability of geared medium-speed installations, using two engines of the same range but perhaps differing in number of cylinders, to almost any combination of propulsion and auxiliary power requirements; the requirements imposed on clutches, a shaft brake (if fitted), or a controllable-pitch propeller by the need for acceptable "crash-stop" manœuvring characteristics; the savings which can be achieved in medium-speed installations by (*a*) waste-heat recovery, and (*b*) choosing a favourable (low) propeller speed.

A combined plant, in which the two or more engines have the same standard components and run on heavy fuel, for main and auxiliary power both at sea and in port, is attractive as regards running costs, maintenance, and availability. Plans of two possible aft engine rooms of this kind are given. One is a "father and son" installation of 11,000 b.h.p. (total); here a large and a small engine drive the propeller through clutches and gearing at their forward ends (this location is convenient for power take-offs), and an additional clutch and an alternator are interposed between the smaller engine and the gear. The other scheme has twin engines (total 13,500 b.h.p.) with clutches and gearing aft of them; each engine also drives an alternator through a subsidiary gear train on the input side of its clutch. It is pointed out that in such cases the engine builder is responsible for the entire plant, including all ancillaries and control and monitoring systems.

Information on the transient characteristics of fixed-pitch merchant-ship propellers during a crash stop is still limited; model tests were therefore sponsored at the N.P.L. In these, various model propellers were fitted to a typical hull which was towed at constant speed with the propeller turning at constant astern r.p.m.; this was done over ranges of model speed, r.p.m., blade area ratio, and pitch ratio. Graphs show torque, thrust, and deceleration curves during propeller reversal from an ahead speed of 15 knots, for a vessel of length b.p. 335 ft, loaded displacement 8,260 tons at 25·5 ft draught, and C_b 0·64; she has twin geared engines, providing 4,700 s.h.p. at 155 r.p.m. and 15·5 knots. The clutches usually fitted in geared medium-speed installations have very limited slip and heat-dissipation capacity, which means that above a certain ship speed (6 knots for the vessel mentioned) no clutch engagement can be allowed. This can result in unacceptable stopping time and head reach (8 min and 22 ship lengths in the example). The curves therefore show the effect of: (*a*) Using a shaft brake (500 h.p. equivalent) to stop the propeller and thereby increase its resistance to the ship's forward motion; the brake capacity must at all times overcome the torque of the freewheeling

propeller, but little is gained by choosing a much larger capacity. The improvement in stopping performance is substantial (6 min and 17·5 ship lengths) but still not sufficient. (*b*) Using clutches which can withstand the slip and energy dissipation involved in re-engagement of the reversed engines as soon as the torque in the shaft has dropped to zero. A graph shows how the slip torque and the power to be dissipated (750 h.p. peak) vary during the manœuvre as a function of percentage slip. This scheme is very satisfactory (3·25 min, 8·5 ship lengths).

With scheme (*b*), the re-engaged engines must be able to satisfy the propeller torque demand at all subsequent stages of the manœuvre; this may necessitate an increase in their idling speed. The point is illustrated by reference to propeller-torque/r.p.m. and power-dissipation/percentage-slip curves calculated for two ships, designed for 16½ and 18 knots respectively with 9,875 s.h.p. at 119 r.p.m.; shaft brakes are used in conjunction with slipping clutches, but the idling speed in the 18-knot ship has to be raised from 200 to 262 (i.e. half of rated) r.p.m. Further curves show that the situation is aggravated by a relatively low propeller speed rating (105 r.p.m.) such as is often advocated on grounds of propulsive efficiency. Failure to meet the torque demand results in stalling or greatly increased slip time, unless clutch engagement is delayed by a control system.

The choice between clutches only and clutches assisted by a shaft brake is discussed in relation to a larger ship (427 ft o.a., loaded displacement 26,000 tons at 26 ft draught, C_b 0·67; 13,160 s.h.p. for 18 knots). Curves analogous to those already mentioned are given. It is found that a shaft brake gives only a slight manœuvring advantage over slipping clutches each capable of dissipating 2,800 h.p. (peak) for 3 sec and 1,200 h.p. (mean) for 10 sec. The latter scheme is therefore recommended on grounds of compactness and simplicity.

Several clutch/coupling and gear manufacturers are co-operating in the production of combined units; these will eliminate quill shafts and associated torsional-vibration problems.

Emergency stopping with a c.p. propeller is also considered, and illustrated by transient machinery-performance curves recorded in a ship of 210-ft o.a. with a direct-coupled engine giving 1,600 b.h.p. at 300 r.p.m. for 13·5 knots. With geared engines, the usual type of clutch would be adequate for c.p. installations.

Whatever the manœuvring method, the control system must incorporate safeguards to prevent machinery being overloaded, stalled, or turned in the wrong direction; this matter is discussed. It is advantageous for the total engine inertia to be at least equal to that of the propeller, gearing, etc.

Increasing numbers of medium-speed installations are being equipped to run on 3,500-sec fuel at all times, which results in substantial cost savings. Waste-heat recovery is economically justified if the machinery spends most of its working life at over 50% of the service load rating; medium-speed installations could in many cases meet this condition in port as well as at sea. Perhaps the most readily applicable exhaust-heat recovery system consists of a gas-heated economiser and an oil-fired unit in which the steam is flashed off; it is advantageous to use the jacket water in a feed heater. Some figures are quoted to illustrate the savings achievable with waste-heat electric power generation.

Finally, calculated data are presented to show the benefits of using a

larger, slower-running propeller in a ship of about 14,000 tons d.w. with a service speed of about 14 knots. It is assumed that these benefits are realised as a reduction in machinery power rather than as an increase in ship speed. Propeller and propulsion data are tabulated, and power/speed curves given, for several values of rated r.p.m. from 150 (requiring 5,500 b.h.p.) to 90 (requiring 5,000 b.h.p.). Assuming that the 3,500-sec fuel costs £5 per ton, and making other reasonable assumptions, the annual fuel-bill saving is £2,260, which is of the same order as that attainable by waste-heat recovery.

26,914 **Steam versus Diesel—up to 3,000 h.p.** BARCLAY, C. *Shipp. World & Shipb.*, **162** (1968), p. 1062 (July) [3 pp., 2 phot.]

The Author argues that there is still a case for propulsion of the smaller cargo vessels by reciprocating steam plants using engines and boilers of modern design. This applies especially in operating areas where Diesel installations encounter manning and maintenance problems.

FUELS AND COMBUSTION TECHNOLOGY

26,915 **Specific Heat of Flue Gases.** *Works Engineering & Factory Services*, **63** (1968), p. 14 (Apr.) [2 pp., 3 tab., 1 graph]

HEAT TRANSFER AND INSULATION

26,916 **Heat Transfer to a Fluid Flowing Inside a Pipe Rotating about its Longitudinal Axis.** CANNON, J. N., and KAYS, W. M. *A.S.M.E., Paper No. 67-HT-82, presented* 6–9 Aug. 1967 [5 pp., 2 ref., 1 tab., 4 graphs, 4 diag., 2 phot.]

The effects of tube rotation on heat transfer to a fluid flowing inside a tube are examined. The most marked influence is on the transition from laminar to turbulent flow; there are lesser effects in the laminar region, but no measurable effects once the flow has become fully turbulent. Heat-transfer data obtained experimentally are presented for a wide range of through-flow rotational Reynolds numbers. Examination of the flow in a visualisation apparatus showed that tube rotation tends to stabilise laminar flow, and can cause an already turbulent flow to revert to laminar flow. When the tube is rotating, the transition from laminar to turbulent flow is characterised by a distinct "burst of turbulence" phenomenon, photographs of which are presented.

AUXILIARY EQUIPMENT AND MACHINERY

26,917 **Shaft-Driven [A.C.] Generators with Controllable-Pitch Propellers.** FARBERG, L. *Motor Ship*, **49** (1968), p. 19 (Apr.) [2 pp., 1 graph, 1 diag.]

The controllable-pitch propeller offers an easy solution to the frequency control of shaft-driven A.C. generators because of its ability to effect manœuvring and ship-speed changes without changing shaft r.p.m.

The Author describes four methods of arranging such shaft-driven alternators. Where medium-speed machinery driving the propeller via a reduction gear is employed, the gearbox can be fitted with power take-off shafts, the shaft speed of which can be selected to suit the alternator drive. These shafts can be directly meshed to the main gearwheel and will then

run with fixed position relative to the propeller; their alternators can be paralleled with each other on common bus bars. The clutches for connection/disconnection of the engines are mounted on the input side of the gear. In an alternative arrangement, each alternator is directly connected to its Diesel through the gear, and the engine clutches are arranged so that this connection is not affected when the engine is disconnected from the main drive. Thus the alternator can be run when the propeller shaft is stopped. However, two such alternators cannot be paralleled, as the relative position of the rotors varies each time they are engaged.

Where direct-coupled low-speed Diesels are installed, the simplest scheme is to couple the alternator directly to the forward end of the main engine, but, as the r.p.m. cannot be selected, the alternator has to be comparatively large. This drawback is alleviated if part of the intermediate shaft forms the rotor and the stator is mounted round the shaft line. A more suitable alternator speed can be achieved through a simple speed-increasing gear, thus reducing the size of the alternator; a flexible coupling should be incorporated. A typical scheme of this kind is shown in a diagram. This method will probably be favoured in future ships with direct-coupled Diesels.

The control system for such installations will normally be designed for two modes, viz. one in which r.p.m. is kept constant at the rated value and all manœuvring carried out with pitch control, and the other for manœuvring below 50% of full power, where a "combinator" control system giving simultaneous control of pitch and r.p.m. is preferable to ensure smooth manœuvring. However, a compromise design of propeller could be produced to give good performance over a wide power range while running at rated r.p.m.

Where twin-engine drive to a single screw is employed, automatic disconnection and re-connection of one of the engines at 50% power is possible through a power (torque) sensing device. The disconnected engine can be maintained at rated r.p.m. for the alternator drive while the other is controlled by the combinator.

With automatic load control of the propeller, primarily intended to protect the propulsion engines against overloading and thereby enable them to be run at higher average powers, it is possible to switch from main-engine to auxiliary-engine alternators or vice versa without a blackout period.

A shaft-driven alternator can also be used for power supply to a bow thruster by direct connection in a separate loop which can tolerate high starting currents.

The Author concludes that, due to the cost savings associated with power generation by main rather than auxiliary engines, the shaft-driven alternator system should pay back installation costs within a short operating period.

26,918 **Theoretical Analysis of the Thermal and Hydraulic Characteristics of Sea-Water Systems for Low-Speed Diesel Installations** (in Russian). MASLOV, V. V. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, **86** (1967), p. 11 [13 pp., 1 tab., 11 graphs, 2 diag.]

After explaining the basic theory, the Author makes a detailed com-

parison between the characteristics of possible series and parallel sea-water heat-exchanger systems for Burmeister & Wain low-speed Diesel installations of various powers.

Over a wide range of powers, the series layout appears at first sight to be more rational. However, this is only true if a full flow of sea water is fed through the air cooler, and if the main pipe is not restricted. Use of the series layout without such restriction makes the standardisation of pumps impossible, since the fresh-water and sea-water systems then require pumps of different characteristics. Thus the parallel system is in fact more rational, because it allows the standardisation of pumps. The parallel system must be employed where the engine pistons and cylinders have a common fresh-water cooling system, and can also be used where the engine has separate cooling of pistons and cylinders.

A tabular comparison is made of calculated data for a series system (with and without restriction) and a parallel system for a B. & W. engine of 8,750 b.h.p., and of actual data for the system in the motor ship *Bezhitsa* (the performance of which is shown to be far from the optimum).

The analysis should aid the designer in approaching the problem of choosing sea-water system layout, pump output, and heat-exchange surface area from the standpoint of minimum pump power, which depends on the method of water distribution in the main and auxiliary machinery, pipe resistance, and the intensity of heat transfer in the main heat exchangers.

26,919 **Energy Transfer in Centrifugal Pumps.** YEDIDIAH, S. *A.S.M.E., Paper No. 67-FE-22, presented* 8–11 May 1967 [11 pp., 14 ref., 17 graphs, 6 diag.]

The mechanics of energy transfer from the impeller blades to the liquid in centrifugal pumps are discussed. Some relations between the shape of the blades, the pressure distribution, and several performance characteristics are pointed out. An approximate method of evaluating the slip factor is developed, which takes into account the geometry of the impeller. The influence of the shape of the blades on the slope of the flow-rate/head curve and on the cavitation characteristics is also considered. Theoretical considerations are compared with test data.

AUTOMATION, INSTRUMENTS, AND CONTROL DEVICES

26,920 **Engine-Room Automation.** GRORUD, H. F. *Europ. Shipbuild.*, **17** (1968), p. 52 (No. 4) [6½ pp., 3 diag., 6 graphs, 1 phot.]

During the past decade automation has been increasingly used in ship operation, mainly for the automatic control of individual machinery parameters, such as oil or cooling-water temperature, speed and voltage of generating sets, feedwater and combustion in boilers, etc. An important modern development is to treat the ship, or at least all the machinery, as a whole, and to co-ordinate the different individual functions by applying the principles of systems engineering. This calls for new skills in the modern shipyard, covering a wide range of engineering knowledge, close co-operation between departments, and sometimes the re-design of machinery units to adapt them to automatic control.

The Author discusses the application of systems engineering to the problem of the unmanned engine room, and the process variables that should be monitored for this purpose. He takes as an example the Diesel

engine, and shows that, although load and various other operating parameters may be kept constant, large random variations in combustion-space temperatures may occur, possibly owing to faulty atomisers or unsuitable fuels. A suitable system for detecting and controlling over-heating in the combustion space is therefore required; and it should be used as part of a system of scheduled maintenance governed by the thermal and mechanical condition of the engine.

Integrated control systems based on systems engineering require suitable and reliable instrumentation. The components of the monitoring system for the machinery of an automated engine room intended for unmanned operation for periods up to 24 hours are discussed. Such a system may have 120 to 150 sensors for operating alarms and controls, and the various units will probably have to operate under more stringent conditions at sea than in a land-based system. In the Author's opinion the most important factor in the development of suitable instruments is thorough laboratory testing in which the various parameters expected in operating conditions are simulated. The environmental parameters peculiar to marine machinery spaces are temperature, humidity, contaminants (e.g. salt or sulphur), vibration, variations in electrical supply conditions due to the relative instability of ships' electrical power supply systems, spray or splash, and flammability or explosion dangers.

The Author then discusses the following examples of weaknesses or failures in automation systems that have been due either to lack of knowledge of the environment or to the inability of components to with-stand the environment.

1. The sensitivity of temperature switches to variations of environmental temperature may change the set point at which a switch operates by as much as 0·5 C for a change of 1 C in environmental temperature.

2. Bounce or chatter of pressure switches which may be aggravated by vibration.

3. Corrosion of pressure transducers immersed in oil by lubricating-oil additives.

4. Inability of some gaskets to withstand oil.

5. Triggering of alarm and safety circuits by electric " noise ".

6. Burning of some sensors used in the scavenge belt to monitor scavenge-belt fires.

7. Several cases in which a stand-by pump has started up while the other pump is running.

Probably future developments in marine automation will be concerned mainly with information analysis and information handling. This will entail not only the application of computers, but also the more extensive use of continuous-output sensors rather than those of the switching type. The former have the additional advantage that it is possible to check them continuously, and much more easily than the latter.

26,921 **Applied Fluidics: Merits, Limitations, Examples.** STEINER, L. A. *Hydraulic Pneumatic Power*, **14** (1968), p. 401 (July) [7 pp., 1 tab., 5 diag.]

The Author discusses fluidics from the standpoint of its use in control engineering, and considers in broad outline its merits and limitations and the desirable trend of future developments in industrial applications.

The characteristics of fluidics in comparison with other control systems are illustrated in a table which gives, for a simple switching operation, the

switching times and power requirements for mechanical, electro-mechanical, fluidic, and various electronic types of operation. Fluidics comes between the electro-mechanical and electronic modes in both respects. Although the speed of switching is even now only about 1/1,000 of that of electronic controls, it is still rapid enough for many industrial applications. The most promising of these appears to be as the mode of control where it is the natural extension of fluid power, and the development of fluidic elements should be directed towards measurement. Success so far has been achieved mainly in the measurement of dimensions, but attempts are being made to apply fluidics to the measurement of temperature, pressure, and to counting and other purposes.

Two types of multi-diaphragm fluidic elements, designed to operate in the range 3–15 lb/sq in, are illustrated; one is produced in the U.S.S.R. and the other in East Germany. The latter is a relay with two possible signal inputs. The former, also a relay, is being manufactured under licence in Italy in the form of SELP plug-in elements. These are quite small; one of average size has dimensions 1 in × 1 in × 1 in.

Discussing systems design, the Author considers the matching of element to load, the uniformity of component characteristics, and environmental requirements. As regards the last, fluidics can be used in certain conditions in which the use of electronic systems would be accompanied by risk of fire. They are also usually unaffected by mechanical or sonic vibration. They do, however, need a supply of clean air free from moisture, oil, or dust; and in some cases also they consume power when the electric equivalent does not.

In system design it is basically wrong to replace an electric system by a fluidic system component by component. The overall function of the system should be specified, and not the chain of logic elements, and the system designed accordingly.

The following applications of fluidic control systems are described:—

A Swedish missile designed to follow the path prescribed by a pneumatic gyro.

A paper-making machine in which the edge of a moving band is kept in position by fluidic control.

A numerical length indicator which shows the distance travelled by a table to the nearest 0·001 in.

A fluidic inspection machine which automatically inspects the bore of engine cylinders.

A liquid-level control.

A petrol vendor operated by moving-part fluidic devices.

A programmer for the forming of stator coils for large generators, involving bending of copper straps, and brazing, heating, cooling, quenching, and drying operations.

26,922 **The Use of Thyristors in the Phase-Amplitude Compounding Circuits of Ships' Synchronous Alternators** (in Russian). ILIN, G. P., MAKSIMOV, YU. I., and POPOV, O. S. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 87 (1967), p. 49 [8 pp., 3 graphs, 7 diag.]

The Author explains how thyristors can advantageously replace electromagnetic switching elements in excitation circuits of the phase-

amplitude compounding type (for automatic voltage regulation). The suitability of thyristorised circuits of this kind for alternators of 20 to 1,000 kW is illustrated by oscillograms from laboratory tests.

26,923 **The Design of a Reversible Electric Drive for the Hoisting Winches of Floating Cranes, Using a Thyristor Converter and D.C. Motor** (in Russian). SMYSHLAEV, E. I. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 87 (1967), p. 65 [12 pp., 11 ref., 5 graphs, 6 diag.]

DECK MACHINERY AND CARGO HANDLING

26,924 **Automatic Loader Cuts Hold Working.** *Storage, Handling, Distribution*, **10** (1966), p. 87 (Apr.) [1 p., 1 phot.]

Two Austrian firms (Maschinenfabrik Andritz AG, of Graz, and Maschinen U. Transportanlagen GmbH, of Stockerau) have collaborated in the design of the Andritz-MUT ship loader. This can handle 1,800 sacks, or 7,000 cu ft of bulk material, per hour. Its main feature is an enclosed tubular boom 100 ft long, mounted on a fixed or travelling hopper car on the quay. The boom can be slewed (2 × 60°), and raised or lowered, by hydraulic means; it can also be brought into a rest position and supported parallel to the quay. Inside the boom are two belt conveyors—an ordinary one in the part near the quay and a telescoping conveyor in the outer part. This latter conveyor feeds a spiral chute arranged to telescope vertically; the chute is kept in the desired position by automatic hydraulic gear, and projects downwards into the hold of the ship or barge being loaded. Its lower end carries a 360° turntable, to which is attached a short belt conveyor; from this, the stevedores take sacks by hand. If their work is temporarily interrupted, sacks are automatically stacked on the chute up to a predetermined height, after which the whole installation is shut down.

The installation weighs about 90 tons, and requires 25 kW; four stevedores are needed for most loading operations. All motions can be controlled either from the ship's hold by a pendant button switch, or from a control cabin on the hopper car. Safety interlocks are provided. The structural design is said to ensure safe operation even in stormy weather; an anemometer at the end of the boom shuts the plant down when wind velocity reaches the maximum safe value.

26,925 **Tippable Compartment Barges for Bulk Transport.** BAYLEY, G. *Mechanical Handling*, **54** (1967), p. 165 (Apr.) [3 pp., 2 diag., 1 phot.]

If a suitable waterway is available, the cost per ton mile (excluding loading costs) of transporting bulk material is less by barge than by road or rail. The total costs can be further reduced by improved methods of unloading, one of which is provided by a system developed by Strachan & Henshaw Ltd; in this the barge is lifted out of the water and turned over so that its contents are emptied into a receiving hopper. Tipping of small barges has been used before, but the new method enables holds of up to 250 tons capacity to be tipped, and it is particularly suitable for compartment boats or for barges consisting of a train of separate tippable units flexibly interconnected during transport. The method is extremely efficient, and comparatively low in capital costs for discharge

rates above about 300 tons/hr, below which discharge by grab and crane is still probably preferable.

Two versions of this scheme are described, one of which was designed for feeding coal to Ferrybridge power station at an unloading rate of 1,000 tons/hr. This rate does not represent the limit of modern requirements, which may exceed 2,000 tons/hr.

For high capacities, the tipping system is superior to the conventional in respect of number of operators required, capital costs, power consumption, and maintenance.

The largest convenient size of compartment appears to be about 250 tons capacity, and barges of 1,000 or even 1,500 tons could be based on this, if their proportions are suited to the size of waterway. An example is given in which a throughput of coal at 1,500 tons/hr is achieved with a compartment size of 150 tons. Ten barges are required and the estimated cycle time is 6 min per barge, comprising 2 min to tip and return, and 4 min for marshalling.

VIBRATION AND SOUND-PROOFING

(*See also Abstracts No.* 26,908 *and* 26,909)

26,926 **Vibration Investigations on the Turbine Blades of Exhaust-Gas Turbo-chargers** (in German). LUDEWIG, H. *M.T.Z.*, **29** (1968), p. 408 (Oct.) [6 pp., 6 ref., 5 graphs, 6 diag., 6 phot.]

26,927 **Marine Diesel Exhaust Noise: Part I—A Mathematical Model.** JANSSEN, J. H. *Int. Shipbuild. Progress*, **15** (1968), p. 170 (May) [11 pp., 7 ref., 1 tab., 9 graphs, 1 diag.]

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The development of low-speed two-stroke supercharged Diesel engines, particularly those with exhaust-driven turbo-chargers operating on the pulse system, has resulted in a considerable increase in exhaust noise. Such noise particularly affects bridge personnel and functions (sound signals, etc.). The most troublesome is noise mainly involving the low-frequency end of the sound spectrum.

The aims of this research were to develop a method for designing a silencer operating in the low-frequency range of the exhaust sound spectrum, and to facilitate its application in the design stage. The results of the investigation are to be published in five parts, of which this is the first.

This part presents a simple mathematical model of the engine as a sound source, and of its exhaust system as an acoustic wave guide. The scale model technique and pertinent criteria are to be discussed in following parts.

The exhaust-gas turbines are considered as sound sources with high internal impedance. Spectra are computed with the aid of the Fourier analysis of a periodic trapezium gas-flow function in combination with the transfer function of the exhaust system; the transfer function is determined from scale models. The influences, on the sound pressure level, of engine power, number of cylinders, r.p.m., turbine nozzle area, etc., are investigated.

The results of this part show the need for further research. Tentative conclusions are that if the power per cylinder is increased by a factor of two the sound pressure level is increased by 3 dB, but that this level is reduced by at least 24 dB if the crankshaft r.p.m. are reduced by a factor of two.

26,928 **Simulation of the Pitting Corrosion of Hull Plating under Static Loading** (in Russian). CHAPKIS, D. T. *Transactions of the Central Scientific Research Institute of the Merchant Marine, U.S.S.R.*, No. 82 (1967), p. 34 [16 pp., 10 ref., 4 tab., 8 graphs]

The Author describes theoretical and experimental research, the aim of which was to evaluate the effect of pitting corrosion on the static strength of hull plates and thereby arrive at a sounder definition of their effective thicknesses (i.e. the values to be used in strength calculations). Pitting corrosion was simulated by one or more hemispherical holes.

The paper concludes that, unlike new plate, the fictitious effective thickness of plate affected by pitting corrosion depends not only on its geometrical characteristics, but also on its state of deformation; different calculation methods must be used for elastic, elastic-plastic, and plastic deformation. A nomogram is given for finding the effective thickness of a plate having a row of equally-spaced hemispherical holes across a transverse section.

26,929 **Anti-Corrosion Epoxy Coating Review.** KUT, S. *Corrosion Protection*, **14** (1967), p. 10 (Apr.) [5½ pp., 3 diag.]

The two-pack epoxy coating system is essentially a solution of epoxy resin(s) in organic solvent, with flow agent(s), and with or without pigments, to which is added, some 20–30 minutes before use, an amine or polyamide "curing" or "activating" agent, generally also in solvent. Immediately on mixing, a chemical reaction of cross-linking starts, manifested by gradual thickening of the paint, which must therefore be used within a certain time—the pot life—which depends on the prevailing temperature and the particular formulation. On the whole, the coatings with the polyamide curers are rather more flexible and impact-resistant than those using amines.

The Author reviews the chemistry of these coatings and of the curing reaction, and lists the wide range of chemicals, solvents, and other substances to which epoxy coatings of one or other of the many formulations now available are resistant. He then describes a number of typical applications and uses. In the marine field, epoxy coatings find extensive use in the internal linings of cargo and ballast compartments. They can also be used for external protection of the hull. Epoxy zinc-rich paints are also of use as "temporary" primers, to protect steel plates which have been cleaned by shot or abrasive blasting before being stored.

Cleanliness of the surface to be coated is of prime importance with epoxy coatings. For steelwork, abrasive blasting is strongly recommended for the best long-term results. The film thickness is also important; it should be at least 6–7 mils (150–175 microns), but there is no advantage in a much thicker coating.

Epoxy coal-tar coatings were originally devised to overcome the reputedly limited water resistance of the two-pack epoxy coatings—which, incidentally, the Author does not accept. These coal-tar coatings have good resistance to salt water and many chemicals, and are used, *inter alia*, for coating the tanks of crude-oil tankers, for external hull coatings, and for marine and offshore steel structures.

Solvent-free epoxy coatings are now available, which reduce labour

costs in application, since only one coat need be applied. They also avoid dangers due to the solvents of two-pack systems, and simplify ventilation problems in confined spaces. They are, however, more costly than solvent-based epoxies.

Epoxy esters can be used where a moderate degree of chemical or oil resistance is required, but where the conditions are not severe enough to justify the more expensive two-pack epoxy coatings.

OPERATION AND MAINTENANCE

26,930 **Fully-Orbital Tank-Washing Machine from Dasic.** *Motor Ship*, **49** (1968), p. 37 (Apr.) [1 p., 1 diag.]

Dasic Equipment Ltd, a subsidiary of Dasic Chemicals Ltd, have designed and tested a new machine for washing the tanks of large tankers. The machine, designated "Jetstream", is being produced in two sizes capable of delivering 75–80 tons/hr and 30–34 tons/hr of water respectively, with a working pressure of 160 lb/sq in. It is designed for permanent installation.

The machine is fully orbital and is self-contained. A Pelton wheel of 85–90% efficiency drives, through compound gearing, a worm and wheel in the base of the assembly. As this wheel is free on its shaft but integral with the body of the machine, the machine is driven round the shaft at 3 r.p.m. The jets are driven from a bevel-gear arrangement via a further worm and wheel and turn at 1 r.p.m. in the vertical plane. As the body rotates, a star wheel strikes a fixed pin which is arranged to give the wash pattern an advance of 4·3° each revolution. The whole unit is sealed against leakage with PTFE twin seals and should require no maintenance over a four-year period.

A combination of up to ten machines of both sizes, operating in two consecutive groups, will be required to give an almost 100% wash in a wing tank of a 200,000-ton d.w. tanker. About four of the larger machines will be required in a centre tank. Thus, such a tanker would require approximately 150 machines with an installation cost of about £30,000. Ex-works, the two machine sizes cost £160 and £110 respectively.